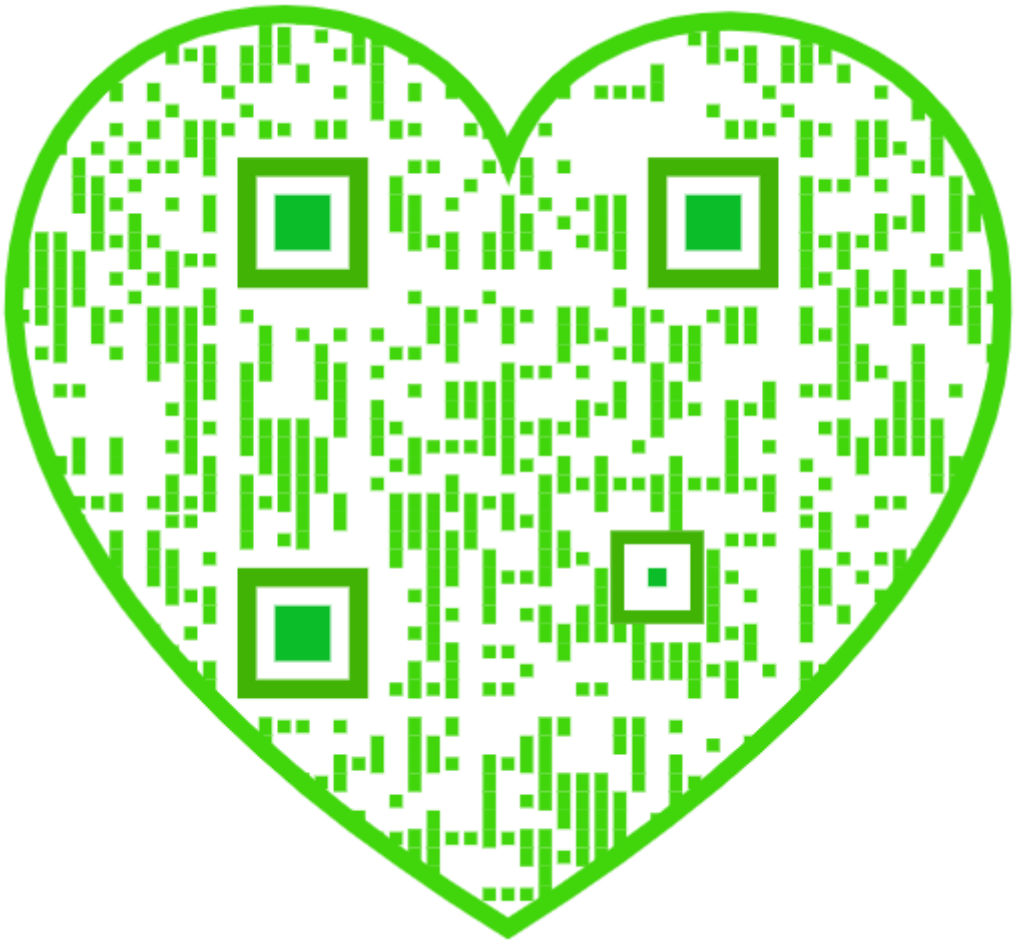

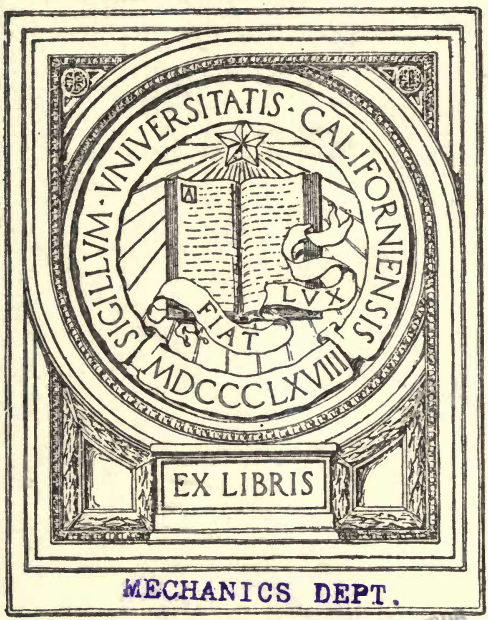

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REFRIGERATION

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THE ELEMENTS
OF
REFRIGERATION

A Text Book for Students, Engineers and Warehousemen

BY

ARTHUR M. GREENE, JR.

Professor of Mechanical Engineering and Dean of the School of Engineering, Princeton University

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PREFACE

The aim of the author in preparing this book has been to bring together in a logical order the necessary data from which to design, construct and operate refrigeration apparatus. He has endeavored to describe the apparatus and then to give the theoretical discussion of the principles on which the action of this apparatus rests. A detailed description of the applications of refrigerating machinery to cold storage and ice making is followed by that of other applications. The author has freely consulted the Transactions of the American Society of Refrigerating Engineers and the bound volumes of Ice and Refrigeration, and has gained much information from these two excellent publications. Much of the text has been developed in teaching this subject for many years. Whenever the work of others has been used, credit has been given. The author is indebted to many writers whose work he has used in the class room and in preparation of his lectures, and to the manufacturers of refrigerating apparatus who have given data for the preparation of this text. The aim has been to make the book complete with the necessary engineering data for problem work without reference to other books.

The author has added a set of problems in the last chapter illustrating most of the computations which must be made in refrigerating work. The problems illustrate the methods by which questions of the engineer may be answered.

The book is intended for the use of upper class men in technical schools, for engineers and those operating refrigerating apparatus. The work presupposes a knowledge of thermodynamics and heat engineering.

The plan of the work has been for a continuous study of the book without any omission. The last two chapters are

553015

intended to give data and methods for actual computations and should be used during the course for problem work. Problems based on the text should be given with the study of the book. These problems should be solved by use of the slide rule.

The author desires to thank his wife, Mary E. Lewis Greene, for the aid she has given in the preparation of the manuscript and in the reading of proof. He desires to thank those authors, publishers and manufacturers who have furnished him with data.

A. M. G., Jr.

SUNNYSLOPE, TROY, N. Y.
September 1, 1916.

TABLE OF CONTENTS

CHAPTER I

PAGE

PHYSICAL PHENOMENA AND INTRODUCTION.—Early methods, evaporation, solution, latent heat, heat of fusion, vaporization under reduced pressure, natural ice, compression machines, air, volatile liquid, general principles 1

CHAPTER II

METHODS OF REFRIGERATION.—Natural ice, different systems of cold storage, refrigerator cars, air machines, open and closed systems, volatile liquid machines, refrigerants, absorption machine, vacuum apparatus, chemical methods..... 8

CHAPTER III

THERMODYNAMICS OF REFRIGERATING APPARATUS.—Air machine, refrigerating effect, cooling, displacement, work, effect clearance, effect friction, incomplete expansion and compression, moisture, vapor machines, temperature-entropy charts, Mollier charts, dry and wet compression, absorption apparatus, problem, multiple effect..... 41

CHAPTER IV

TYPES OF MACHINES AND APPARATUS.—Various machines, cylinders, manipulating valves, pistons, CO_2 machines, SO_2 machines, absorption apparatus, welding, pipes, fittings, condensers, separators, receivers, coolers, vacuum apparatus, binary refrigeration, cooling towers, nozzles, ponds, safety devices..... 109

CHAPTER V

HEAT TRANSFER, INSULATION AND AMOUNT OF HEAT.—Radiation, convection, conduction, transmission, constants, walls, partitions, floors, pipe covering, doors, heat from machines, lights, persons, cold storage data... 182

CHAPTER VI

COLD STORAGE.—Purpose, laws, peculiar features of storage for different articles, layout of warehouse, construction, arrangement of piping,

PAGE

special cold storage, amount of insulation, indirect refrigeration, bunkers, fans, amount of refrigeration, coil surface, pipe sizes, central stations, automatic refrigeration, refrigerator cars, precooling 217

CHAPTER VII

ICE MAKING.—Can system, plate system, apparatus, flooded system, filters, evaporators, raw water system, freezing tanks, expansion coils, curve of consumption, storage 269

CHAPTER VIII

OTHER APPLICATIONS OF REFRIGERATION.—Candy, breweries, blast furnace, auditoriums, rinks, ice cream, shaft sinking, drinking water, chemical works, dairy, creamery, liquid air 312

CHAPTER IX

COSTS OF INSTALLATION AND OPERATING COSTS.—Land, buildings, machinery, supplies, dimensions of apparatus, power and performance of plants, labor costs, load factors, operating costs, ice data, car data, ice cream data, ammonia, carbon dioxide, sulphur dioxide, testing apparatus, results of tests 343

CHAPTER X

PROBLEMS.—Insulation, space, refrigeration for rooms, coil surface, pipe length, velocity of brine, bunker coil, fan size, brine main and pump, ammonia main, plate plant, refrigeration of plant, ice storage, cost of pumping, evaporator, filter, compressor, power, condenser, multiple effect, water cooling tower, blast furnace refrigeration, test computations. 407

ELEMENTS OF REFRIGERATION

CHAPTER I

PHYSICAL PHENOMENA AND INTRODUCTION

THE practice of cooling bodies below the temperature of the surrounding atmosphere has been followed for ages. This has been done by the **evaporation** of a liquid as is the practice in Mexico and other warm climates, where the liquid to be cooled is hung in porous vessels. The evaporation of the liquid which percolates through to the outside cools that remaining inside. In India, it is stated, evaporation from the surface of shallow porous vessels even causes a film of ice to form. The **solution** in water of a salt like saltpetre, or the mixture of snow or ice and saltpetre, has been used for centuries to abstract heat and cool the liquid resulting, or anything that was immersed in it.

The first method was applied about one hundred and fifty years ago in a way differing from that of the ancients. It was then found that **evaporation** of the liquid would occur if the pressure were removed, particularly if the liquid were ether or some other highly volatile liquid. This evaporation would occur at such a low temperature that ice would rapidly form on the surface of the vessel containing the boiling liquid if the vessel were placed in water. It was also found that if the vapor arising from the evaporation of the liquid were compressed to a higher pressure than that at which evaporation took place, it could be condensed again by water at ordinary temperature, and the process repeated.

The property of the substances utilized in these illustrations

is the property of **latent heat.** When a body changes its state, a certain amount of energy must be absorbed by that body to bring about this changed state. To change a body from a **solid** in which the form and volume are fixed and the condition of the molecules is such that their orbits are fixed, into a **liquid** in which the volume but not the form is fixed, or the molecules have orbits which have more freedom, requires the addition of energy. The name **heat energy,** or **heat,** is applied to this. Energy is required to change a body from the liquid state to the vapor state in which the form and volume are not fixed, since the molecules have free paths. The molecules are so far apart that molecular attraction has been broken down. The energy required in the case of the **fusion** of a solid or the **evaporation** of a liquid is used to overcome molecular energy of attraction, and for that reason it is **potential** in form within the body. It is not used up in increasing the kinetic energy of the particles of the body, and hence there is no change of temperature during these additions, and the heat is called **latent heat.**

In general, if heat be continuously added to a solid while the pressure remains constant, its temperature will rise until the **point of melting or fusion** is reached and then the temperature will remain constant until the solid is changed to a liquid. The temperature of the liquid will then continue to rise with the addition of heat until the **boiling-point** is reached, at which point the temperature will remain constant while the heat is added, although the liquid will be changing to a vapor. The further addition of heat will increase the temperature of the vapor.

The previous operations were supposed to take place at constant pressure, because to every pressure there corresponds a temperature of fusion and a temperature of vaporization. These are fixed for definite pressures, and at these pressures and temperatures the amount of heat to fuse 1 lb. of substance, **the heat of fusion,** and the heat required to vaporize 1 lb. of liquid, **the heat of vaporization,** are fixed. Should the pressure change, the temperature of these actions would change.

In the ancient way of producing cool liquids, the evaporation which occurred at the outside of the vessel required heat, and this was largely supplied from the liquid within. The liquid was cooled by the removal of heat. If this removal of heat cools the liquid to its freezing-point (fusion-point of solid), any further evaporation of the liquid from the surface of the vessel would remove heat from the liquid and cause some of it to solidify, forming ice if the liquid were water. In the case of salt being dissolved, this same kind of energy is needed. In this case it is called the **heat of solution.** To change the condition of the molecules of the salt so that the molecular forces are overcome, energy is applied, and as this energy comes from the liquid, its temperature is lowered.

In the case of the **vaporization of a liquid under reduced pressure,** the object of this reduction is to permit the evaporation at such a low temperature that heat may be removed from surrounding objects of low temperatures. Water boils at 212° F., but if the pressure were reduced to $\frac{1}{20}$ lb. the temperature of boiling would be less than 32° F. and with the evaporation of some liquid, ice could form. Of course, it must be remembered that the evaporation of a liquid can take place only if heat is added to it at the boiling-point. If the liquid is at the boiling-point and there is nothing from which heat can be abstracted, nothing can happen. If it is in contact with substances at temperatures below the boiling-point nothing will happen. For this reason the pressure on the lower side must be such that the boiling temperature is below the temperature of the body from which it is to abstract heat, and when evaporated, the pressure must be raised to a point at which the boiling temperature will be above that of the substances used to abstract heat. In this latter condition the substances will abstract heat from the vapor and condense it.

These methods have been used for years to obtain cool water, to preserve foods and for other purposes. In many places, however, this preservation was carried on by the use of **natural ice** harvested in the winter and stored until needed in warm weather.

It was about the middle of the last century that the Carré Brothers produced **commercial machines for the freezing of water.** Both machines operated to remove heat by vaporization of a volatile fluid, Edmund Carré evaporating water vapor at very low pressures and Ferdinand Carré evaporating liquid anhydrous ammonia. These machines were not used to produce large quantities of ice, but they produced commercial quantities.

The **compression type** of machine introduced in 1835 by Perkins was further developed by Twining, who took out his English patent in 1850 and his U. S. patent in 1853. In this machine a volatile liquid such as ether, carbon disulphide or sulphur dioxide is allowed to flow through a throttle valve into a region of such low pressure that the boiling temperature of the liquid is low. This liquid will boil by the abstraction of heat from the substance around the walls of the chamber in which it is placed. The pressure is maintained at a low point by the suction of a compressor which removes the vapor as it is formed and compresses it to a higher pressure. This pressure is high enough to give a temperature of boiling or liquefaction higher than that of a water supply. The water will remove heat from the vapor and cause its liquefaction. The liquid is then passed through the cycle again. In this system the liquid and its vapor are kept separate from everything else by being contained in a closed system. Such a machine produced commercial quantities of ice.

There is one other method of abstracting heat, which has been used for some time. This is the **compressed-air method.** If air is compressed rapidly, its temperature is increased, due to the work which has been done upon it. This air may be cooled to its original temperature by being passed through pipes over which water is allowed to flow, and if this high-pressure air is permitted to drive a piston and do work, the work done will cause a decrease in temperature, so that the expanded air will be so cold that it will abstract heat from a space or room through which it may be passed in pipes or in the open on its way to the suction of the compressor. In this case

the heat abstracted in the **refrigerator** and that equal to the difference between the works of the compressor and of the expander are taken up by the cooling water. In this machine the compressor and expander work on the same shaft.

In all mechanical refrigerating machines the **working substance is placed in such a condition that it will abstract heat from the material of low temperature and after this absorption it is placed in such a condition that it will give up this heat and that added to operate the process, to a water supply at a higher temperature than that of the refrigerator space.** This is the **general principle of all refrigerating machines.**

In the middle of the last century a development of the western part of the United States took place, and with it arose a desire to ship fruits from the central parts of the country to the East. In 1866 refrigerated boxes holding 200 quart baskets of strawberries and 100 lbs. of ice were built. These weighed complete 600 lbs. They proved that fruit could be shipped if kept cool. This was done by Parker Earle. In 1868 Davis of Detroit proposed to insulate cars to handle beef and fish, and in 1872 there were successful experiments. This was the beginning of the refrigerated car industry, which has so extended that in 1910 there were over 130,000 cars in the United States, although only a little over 1000 in Europe.

The original refrigeration and even a large amount of modern refrigeration have been accomplished by ice. The machines for the manufacture of the so-called **artificial ice,** or better **manufactured ice,** have made possible the refrigerating of stores or other houses by the use of this apparatus without the employment of ice. In these **cold-storage warehouses** the volatile liquid may be passed through pipes in the various rooms, from which it abstracts heat and vaporizes, or the evaporation of the liquid may abstract heat from a **strong brine** of a very low freezing-point. This cold brine is pumped through the rooms, removing heat. This latter method is spoken of as **the cool-brine system of refrigeration,** while the former is called **the direct-expansion system.** The method of mechanical refrigeration has made it possible to care for storage in warm countries

at a distance from ice fields. It has permitted the refrigeration of certain portions of vessels during long voyages. It has also led to the possibility of the cold storage of food products. In 1905 it was stated that the value of food products in cold storage in the United States amounted to over $200,000,000, and the investment in refrigerating apparatus amounted to over $100,000,000.

The first long-distance shipment of meats in refrigerators on shipboard was in 1873, but it was unsuccessful. In 1875 successful shipments were made from America to England, and in 1880 Australia shipped meat to England. These shipments have so grown that in 1910 the United Kingdom imported nearly 13,000,000 carcasses of lamb and mutton and over 4,000,000 quarters of beef from South America, New Zealand and Australia. In 1904 the United Kingdom paid $45,000,000 for fruit, of which one-ninth came from the United States. In 1910 there were more than 800 vessels equipped for the transportation of food products in cold storage.

From the above the original importance of this mechanical refrigeration is seen, but with its development further applications have been made and at the present time its use enters into many industries.

Cut flowers are kept for a considerable time, and even trees may be held dormant for weeks to prevent budding before transplanting in the spring. Milk and cream may be kept sweet for some time by means of refrigeration. In the manufacture of wine and beer this apparatus is used to prevent the rise of temperature as well as to cool hot liquids. In the refining of oils the apparatus is used for the removal of certain paraffin products. In the ventilation of buildings in warm weather, cooled brine may be employed to cut down the humidity of the air as well as the temperature. This is applied also in metallurgical operations to remove the excess moisture from the air entering a blast furnace, as well as to make the air of uniform quality. In the manufacture of textiles, in the curing of tobacco and in cigar making, in the making of perfumery, in the manufacture of photographic films and other products, as well as

in developing, the use of the refrigerating machine or its product is indispensable. Even in mining and in excavating the refrigerating machine has been applied: in the first case to cool warm excavations, and in the second to freeze a ring of quicksand so that an excavation could be made through this treacherous material. In therapeutics, the value of refrigeration is being seen. Mr. W. T. Robinson has stated that he has known of hay fever patients being relieved by visiting cold-storage warehouses.

CHAPTER II

METHODS OF REFRIGERATION

THE commercial methods of refrigeration or the cooling of materials and spaces are as follows:

1. Natural ice;
2. Air machines;
3. Compression machines using volatile liquids;
4. Absorption machines using volatile liquids;
5. Evaporation;
6. Chemical methods.

In describing these methods and in illustrating them, the endeavor has been made to show certain well-known types of apparatus so that the student may study actual forms of machines. The peculiarities of the apparatus must be noted and studied in the examples chosen, since these are found in most apparatus for this purpose. The examples taken are those known to the author, and represent good practice. There are many machines built of value equal to that of those shown, and in buying machinery comparison must be made between all parts before deciding which machine is the best.

In the application of **natural ice,** which is that employed in the common refrigerator, the ice is used to cool the air in contact with it, and then this air, becoming heavy, drops to the bottom of the refrigerated space, displacing warmer air, which rises to the ice chamber, where it is cooled by the melting of a proper amount of ice. Fig. 1 illustrates the form of refrigerator built by the McCray Company. The ice is introduced on one side of the ice box and the air is circulated downward to the lower part of that side, rising to the provision side of the refrigerator. The walls of the refrigerator are

made of several thicknesses of materials. As shown, it consists of oak, sheathing paper, poplar or some other lumber, sheathing paper, mineral wool, sheathing paper, lumber, felt and opal glass, nine layers in all. This makes a well-insulated box.

In Fig. 2 the **Jackson system of cold storage** is shown. In

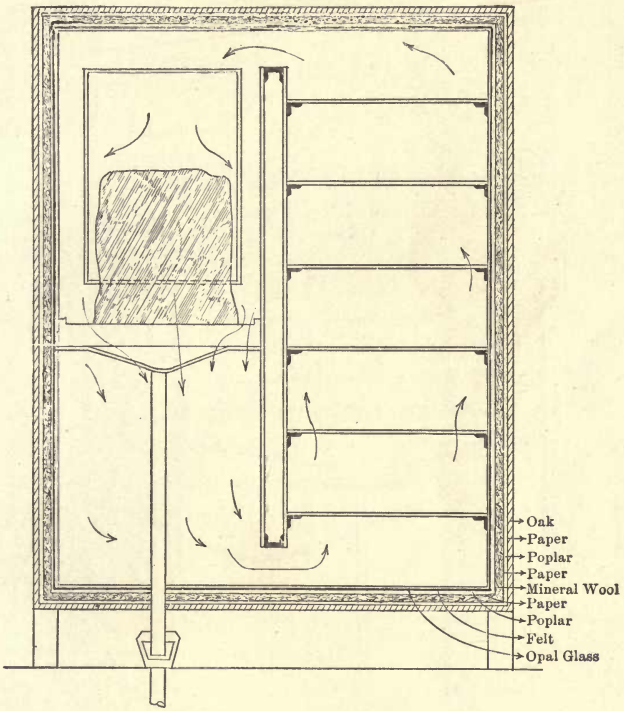


FIG. 1.—McCray Refrigerator.

this the cold air falls around the ice and drops into the cold-storage room, after which, on being heated, it ascends to the ice room. The ice is supported on a slat floor and the drip is caught in the necessary pans, from which it is removed by pipes. The columns are properly protected against this drip. The air leaving the ice chest is saturated with moisture at the temperature of the ice, and as it descends into the warmer portions of the box its moisture capacity is increased, so that

there will not be any deposit of moisture from the air. As the air passes over the goods, there is, if anything, a tendency to take up moisture, and when the air enters the ice chest this

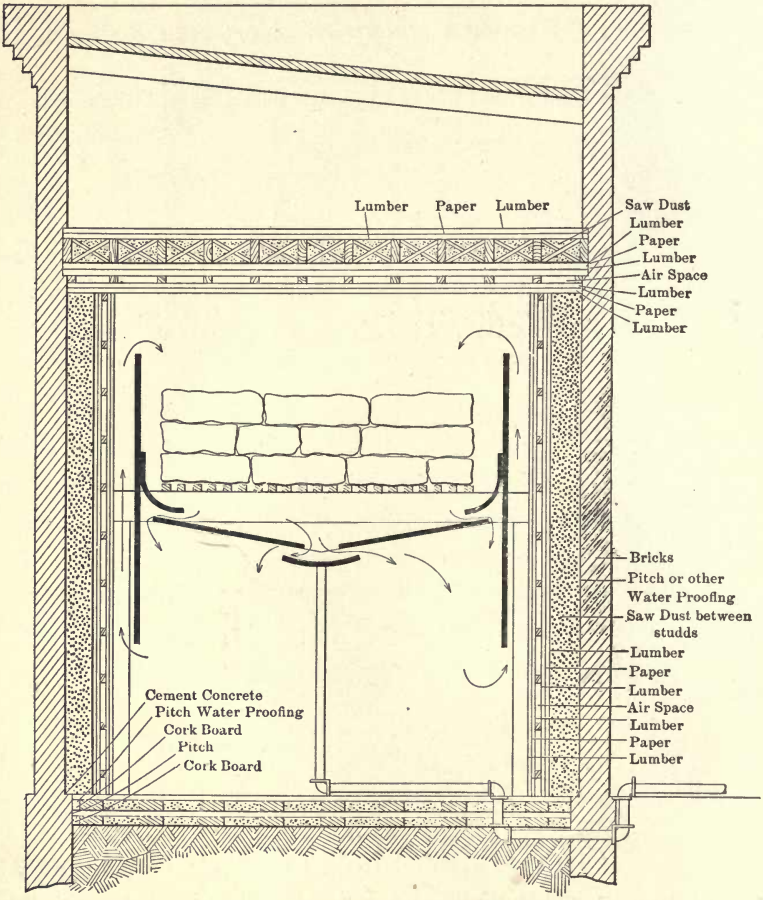


FIG. 2.—Jackson System of Cold Storage.

moisture is removed as the temperature falls. This means additional ice melting. This is not a loss, as the evaporation in the box abstracts heat and this increases the cooling effect at this point, for which, of course, ice is melted later. The

methods of insulating floors, walls, and ceilings are to be examined by the student in the figures shown.

In Fig. 3 the arrangement of the **Dexter system,** in which

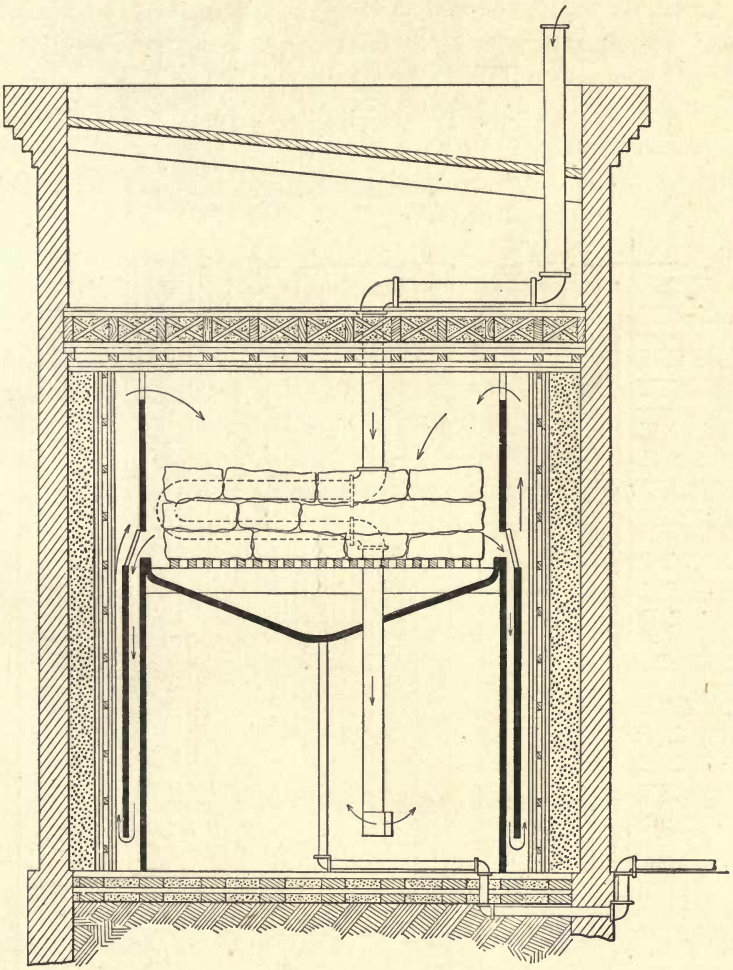


FIG. 3.—Dexter System of Cold Storage.

the air from the ice room does not enter the cold-storage room, is shown. This drawing is self-explanatory. In all these arrangements the water from the melting ice may be taken

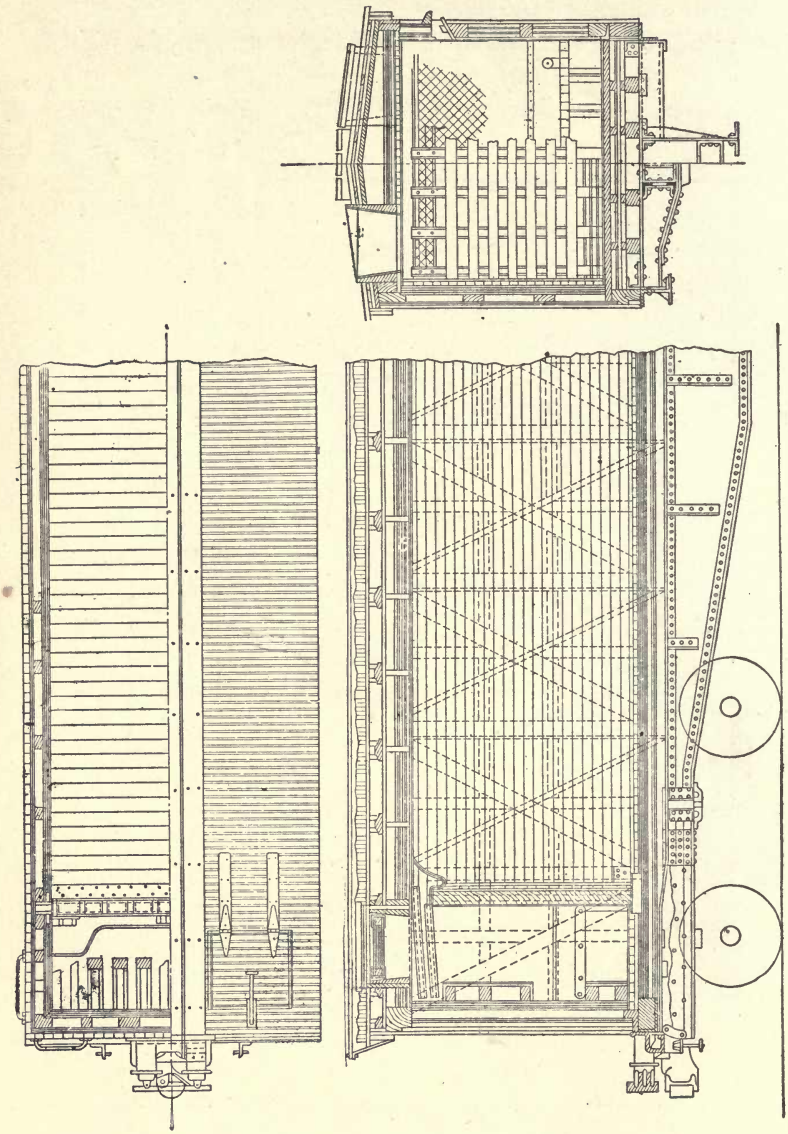


FIG. 4.—Refrigerator Car.

through pipes placed in the cold-storage room. This water is cold and will remove some heat by being warmed to the temperature of the storage room. In this way the apparatus is made more efficient. Fresh air for ventilation may be introduced by a duct leading to the outside through the ice room.

Fig. 4 illustrates the method of cooling **refrigerator cars.** In these ice is introduced at each end of the car, and the circulation of air, in at the top and out at the bottom, cools the air and maintains a low temperature. The refrigerator car shown has been recently built by the American Car and Foundry Co. for the Illinois Central R. R. for their express service. The cars are 50 ft. long and have a capacity of 40 tons. They weigh 75,700 lbs. each. They are supported on steel frames. The car proper is built of yellow-pine framing supported on steel under-framing. The insulation is made up of lumber, insulating material and spaces arranged as shown in the figure. The door section is shown on the right of the cross-section as well as the ice chute on the left. The ice is packed in collapsible compartments at each end of the car, and rests on a rack or support at the bottom. This is the **Bohn collapsible ice box.** From it the air is deflected by curved slats into the refrigerated space. It passes through a screen to prevent the entrance of solid bodies. The rack at the bottom folds up into the end of the car and the slatted front folds up to the roof. The end of the car is protected from the ice by the horizontal strips shown in the cross-section. Ice is charged through the upper doors. Sometimes the ice is broken in a crusher before being introduced and sometimes ice blocks are used. The space for ice is about 3 ft. long, $6\frac{1}{2}$ ft. from ice support to roof, and 8 ft. wide. This would hold about 3 tons of ice at each end. There is an insulating plug in each of the four ice hatches, and the covers are equipped with adjustable latches to give ventilation when needed. The water is drained from the bottom.

In the above installations, temperatures of 36° to 38° may be obtained in warm weather. When lower temperatures are desired, resort must be made to mixtures of salt and ice. The following table, given by T. Bowen in Bulletin 98, U. S. De-

partment of Agriculture, gives the **temperature resulting from mixtures of ice and salt:**

Per cent salt in mixture by weight ..	0	5	10	15	20	25
Temperature of mixture.....	32	27	20	11	1.5	−10

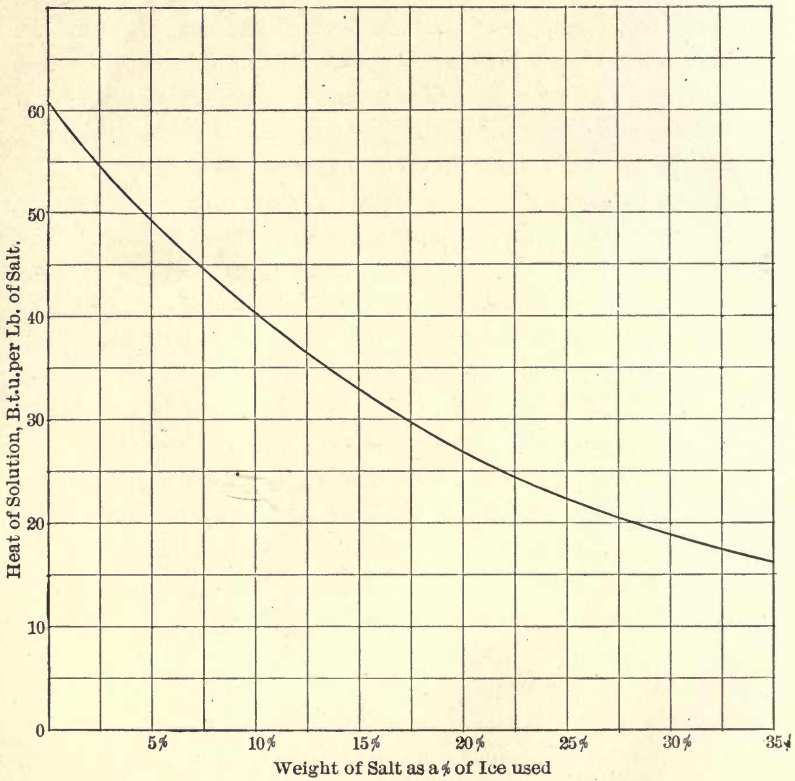


FIG. 5.—Curve of Heat of Fusion of One Pound of Salt for Different Amounts of Salt. (J. T. Bowen.)

The **heat of solution** of the salt varies from 58 B.t.u. to 16 B.t.u., depending on the concentration of the salt. On melting the ice, which requires 143.4 B.t.u., and dissolving the salt, which requires a variable amount, the total heat required will be the sum of that due to the salt solution and the melting of the ice.

For different percentages of salt added to water the **heat of solution** is given by the curve of Fig. 5. Since the heat of solution of salt is less than that of ice, the heat of melting of a mixture of ice and salt per pound of mixture

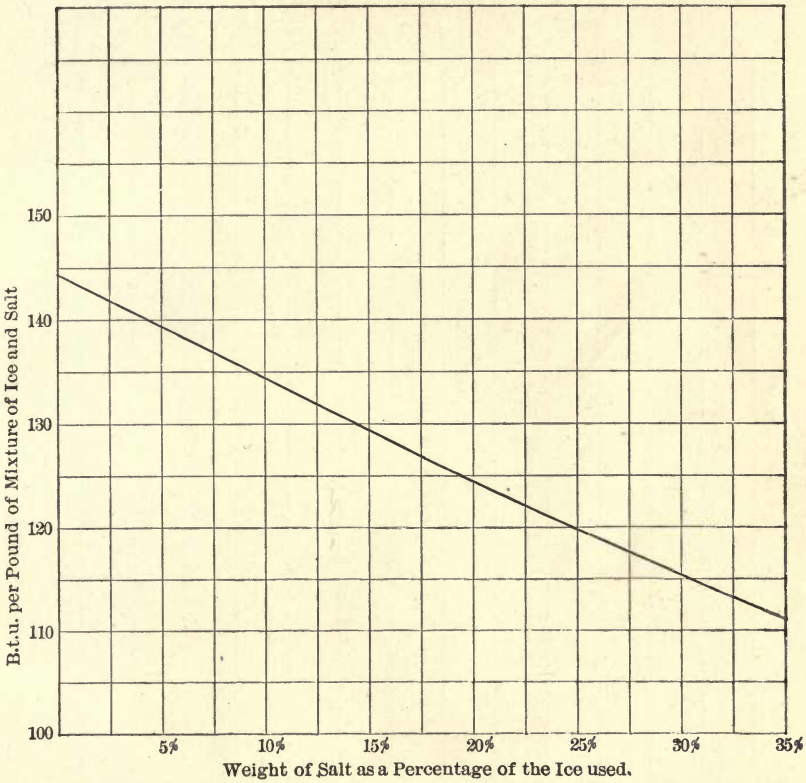


FIG. 6.—Heat of Melting One Pound of Ice and Salt in Different Proportions. (J. T. Bowen.)

decreases as the amount of salt increases. This is shown in Fig. 6.

The specific heat of the salt brine and of ice must be known to make the necessary calculations for the heat removed and the temperature of the mixtures. The **specific heat of the brine** for different percentages of salt is given in Fig. 7.

The **specific heat of ice** at absolute temperature T, is given by

$$c = 0.5057 + 0.001863T + \frac{0.004}{T^2},$$

as determined by Dickinson and Osborne.

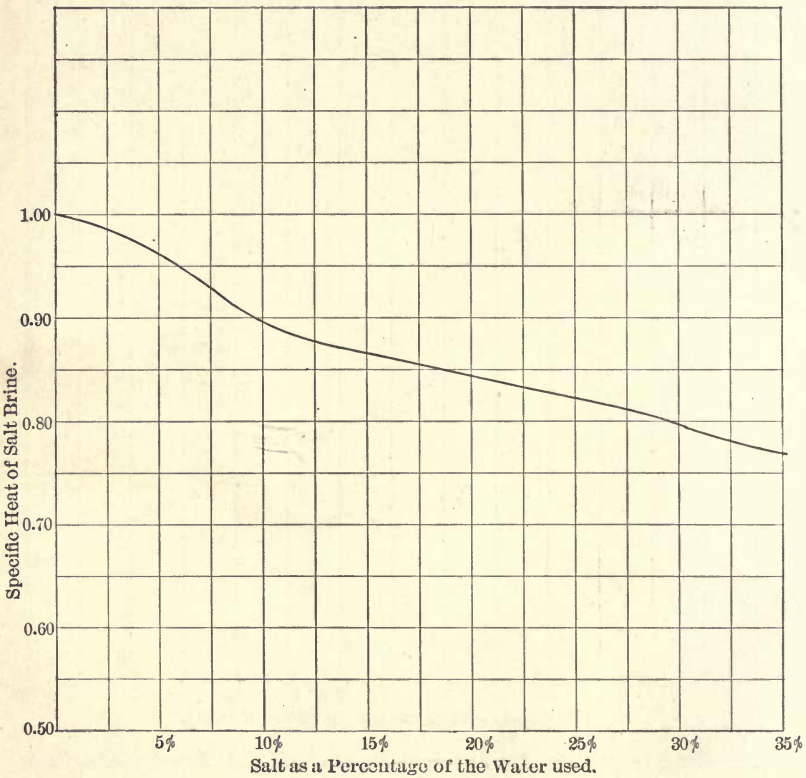


FIG. 7.—Specific Heat of Salt Brine for Different Amounts of Salt.) (Bowen.)

By mixing ice and salt together **low temperatures** may be obtained for chilling or freezing. The ice must be brought into intimate contact with the salt, and for that reason the ice is broken into small pieces. This of course necessitates the ice filling a containing vessel, which is placed in the storage room, although in some cases air is drawn through the mix-

ture. This is difficult, as the mixture sometimes freezes into a solid mass. Then it becomes necessary to increase the

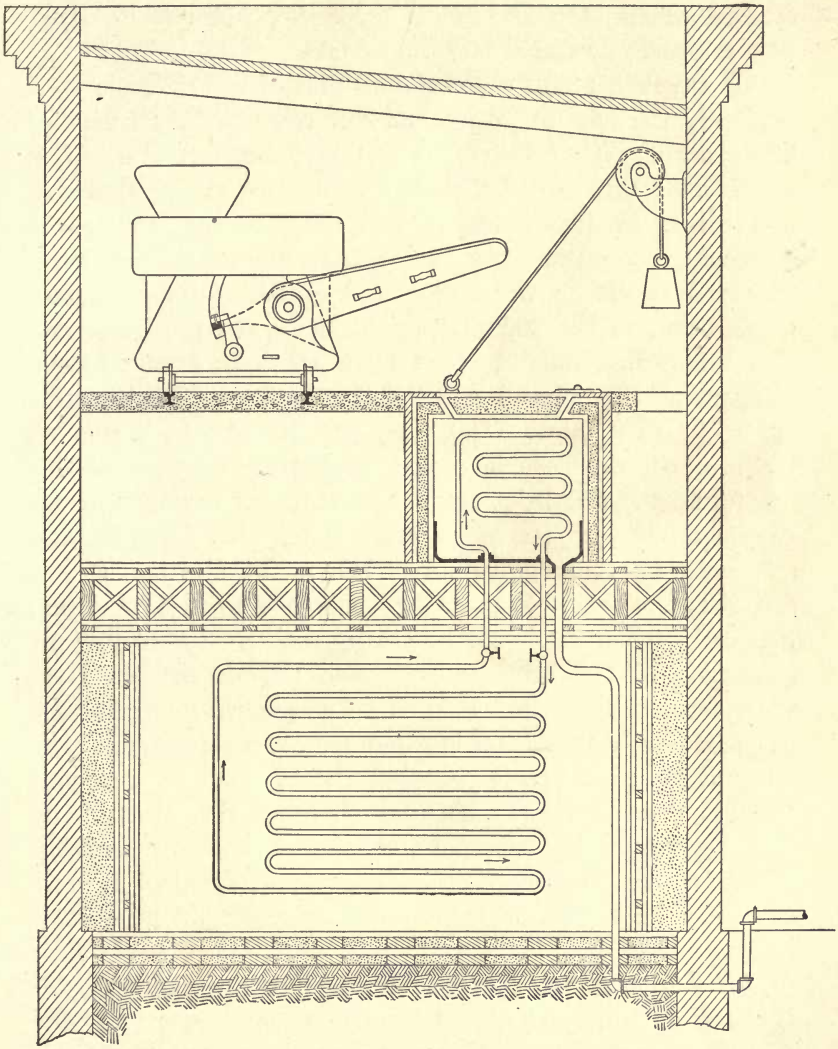


FIG. 8.—Diagram of Cooper Gravity Brine Circulating System.

surface used to refrigerate the room. Certain patented methods have been proposed.

The **Cooper gravity brine circulation system** is shown diagrammatically in Fig. 8. In this ice is broken in a crusher and delivered to the top of the storage house, or it may be crushed at the top after it is delivered. Here it is mixed with salt and introduced into the ice tank.

This tank contains a set of coils filled with brine, and consequently the mixture of ice and salt removes heat from the brine, cooling it to the temperature of the mixture. After this is done there will be no further melting except that due to the heat loss from the tank. When, however, the valves controlling the brine system are opened, the heavy cold brine will tend to fall to the lowest part of the system, bringing warm brine to the top and producing a strong circulation. The brine then removes heat from the refrigerator rooms, being passed through coils on the walls or ceiling. The ice tanks are about 10 ft. high. The lower part is not very active, as the brine in the coils is cooled by the time it reaches this point, and the salt brine from the melting ice cannot take up any more salt, as its temperature is too low. The ice and salt must be thoroughly mixed before introduction, as the salt is apt to cake. When a new charge is to be introduced, the ice in the tank should be stirred with a stick to prevent any caking. The brine formed from the ice melting is of value for cooling in that it is at a low temperature. It may be passed through the refrigerated rooms in pipes, or it may be used at other points. Cooper claims that two men can handle 4 tons of ice per hour in charging this system and that 4 tons per day will cool a storehouse of 40 cars capacity in average summer weather. The amount required in any case may be computed from the heat losses in the storage house.

Air machines are operated in the following manner: Air is compressed in a **cylinder** A from a pressure p_1 to a pressure p_2. The air is discharged from the **compressor** through a set of self-acting **mushroom valves** or by a **slide valve,** and is passed into **cooling coil** B, which is surrounded by water. In this coil the air which has been heated by the compressor is

cooled almost to the temperature of the water, and by this cooling it is reduced in volume and is passed into the **expansion cylinder** *C* This is mounted in **tandem** with the compressor, as in the **Lightfoot machine,** or beside the compressor attached to the same shaft as in the **Allen Dense Air machine.** The air-expansion cylinder is arranged in the same manner as the cylinder of a steam engine. It has a **slide valve** or valve gear, which cuts off the air supply at the proper point and permits **expansion** to occur. This expansion should be **complete** (re-

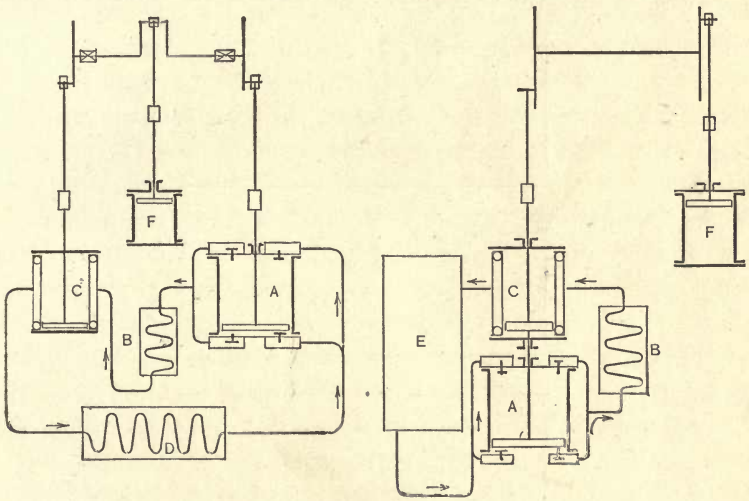


FIG. 9.—Closed and Open Air Refrigerating Machines.

duced to back pressure at the end of expansion). This is accomplished by having the **cut off** occur at the proper point. As this air expands, doing work at the expense of its **intrinsic energy,** its temperature is decreased so that when the exhaust pressure is reached the air may be at some temperature between $-50°$ F. and $-100°$ F. The **temperature** is **fixed** by the amount of expansion. This cold air is now discharged into a **coil** *D* or a **room** *E*, and it removes much heat before it is brought up to the temperature at which it enters the compressor cylinder to repeat the cycle.

The system on the left is known as a **closed system,** while

that on the right in which the air is discharged into the room E is known as an **open system.**

The air occupying less volume in the air expander than it does in the compressor on account of lower temperature, means that there will be less work returned by the expander than that required by the compressor; hence power must be supplied by an external motor of some form.

This may be a steam engine as at F, Fig. 9, or an electric motor may be applied.

To show the work done by indicator cards the compressor

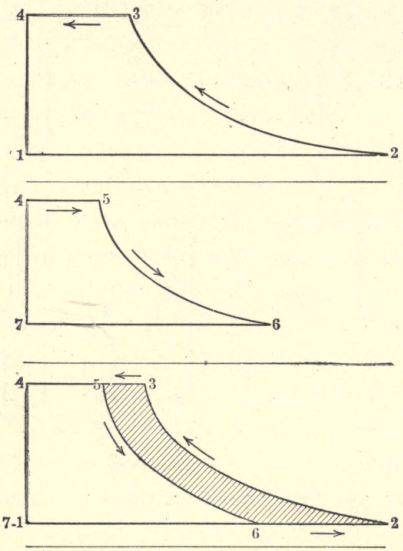


FIG. 10.—Cards from Air Refrigerating Machines.

and expander cards are shown in Fig. 10, assuming zero clearance. This may be assumed, since clearance does not affect the work of a card. These cards may be superimposed and the area 2356 shows the net work which must be supplied if friction be disregarded. This is the work that the motor must supply. The real work, of course, if f is the percentage friction, is

$$\text{Net work} = \left(\frac{100+f}{100}\right)\text{area } 1234 - \left(\frac{100-f}{100}\right)\text{area } 4567.$$

In this machine, **air, by compression, is put into such a condition that the water supply will abstract heat and then, by expansion, it is put into such a condition that it will abstract heat from a place of low temperature.**

The **advantages** claimed **for these machines** are: the use of no chemical which might lead to explosions or loss of life due to accidental escape of gas; the possibility of very low temperatures; simple construction and the accessibility of all parts.

The first air machine was designed by Gorrie in 1849. Kirk designed one in 1863. In 1877 the Bell-Coleman improvements made the machine practical, and the application of this machine to ocean steamships made possible the cold-storage shipments of meat. The machine was improved by a number of later inventors.

There are few air machines used to-day, owing to the greater efficiency of types of apparatus using other working substances. But efficiency is not always the criterion by which to judge of the advisability of using a certain form of machine. Reliability, ease of operation, small maintenance cost, and absence of poisonous substances may be important factors to consider in making a selection. For such reasons air machines are still in use. One of the most common forms of air machines used in the United States is the **Allen Dense Air Machine,** shown in Fig. 11. In this the three cylinders, steam, compressor and expander, are placed beside each other, and are connected to three cranks of a common shaft. The **cylinder** *A* in the front of the figure is the **expansion cylinder**, the second is the **compressor cylinder,** and the back cylinder is that of the **steam engine.** Between the compressor and expander is seen the plunger of a small **air pump** used to maintain the air pressure in the system and make up for any leaks. This is driven by an extension from the cross-head of the compressor. A similar extension on the other side of this cross-head drives a plunger of the **circulating pump** which forces water through the cooling chamber *B* placed on top of the machine. Two **eccentrics** on the shaft control the various valves. The **pipe**

C takes the cool compressed air from the coils in the water-cooler to the expansion cylinder *A*, and after expanding, the

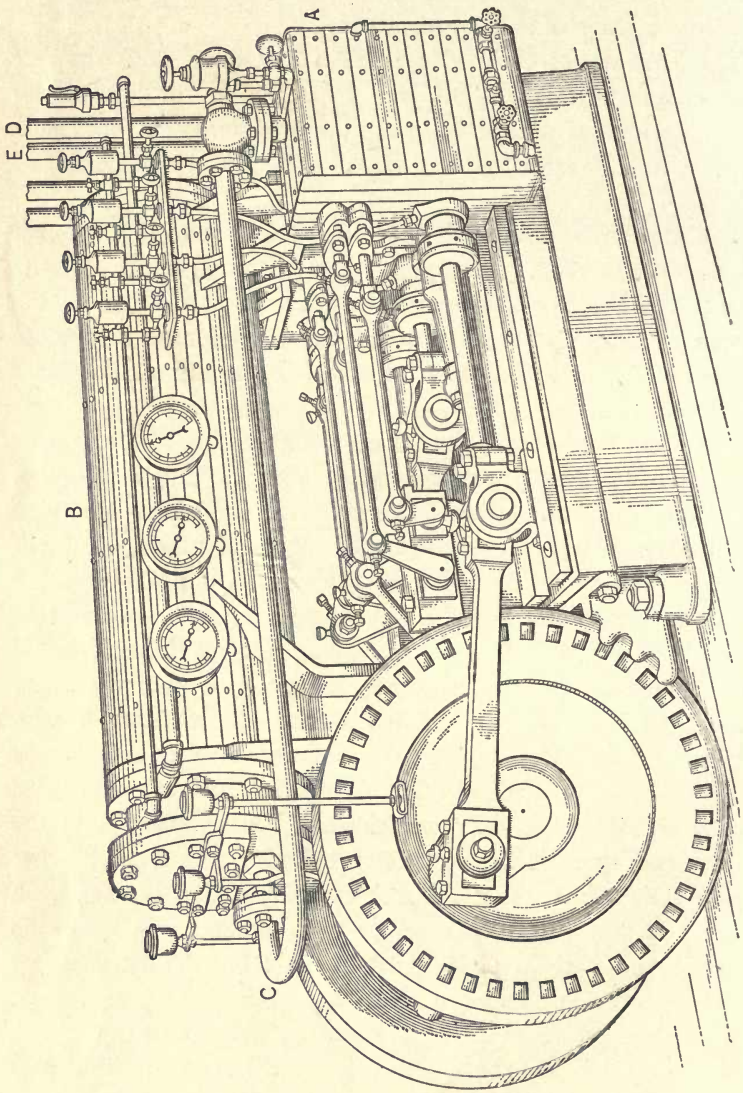


FIG. 11.—Allen Dense Air Machine.

air is passed through pipe *D* to the **refrigerator,** and then returned to the compressor through *E*.

This apparatus is known as a dense air machine. The air is at 60 to 70 lbs. gauge pressure on the low-pressure side,

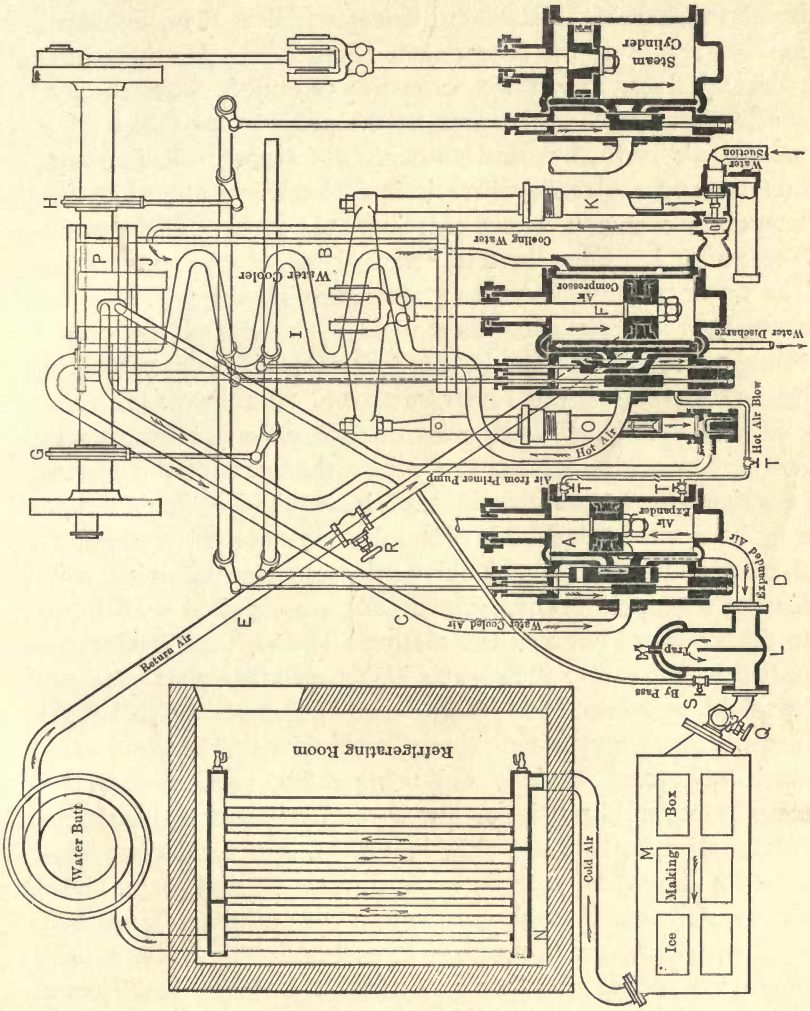


FIG. 12.—Allen Dense Air Machine of H. B. Roelker.

the high pressure ranging from 210 to 240 lbs. As will be seen later, the refrigerating effect is due to the ratio of these two pressures and not to the absolute value of each. By using a high pressure on the lower side, the displacement of

the cylinders for a given amount of refrigeration is materially decreased.

The diagrammatic arrangement of the machine is shown by the maker, H. B. Roelker (Leicester Allen, 1879, inventor) in Fig. 12. *F* is the compressor cylinder, in which a special pair of valves, driven by eccentrics *G* and *H* through **rock shafts,** give ample outlet area at the proper time. The compressed air is then passed through the **copper coil** *I*, placed in the **water-cooler** *B*, where it is cooled down almost to the temperature of the water entering at *J* from the **pump** *K*. The pump is driven from the cross-head of the compressor. The water used in the cooler passes through the **jacket** of the compressor before being discharged. The cooled air passes through the pipe *C* to the expander *A*, which is controlled by a **riding cut-off valve gear** as shown. After expansion the cold air leaves by *D* and passes through an **oil and snow trap** *L*, entering an **ice-making box** *M*, or that part of the system in which the lowest temperature is required. The ice box may be a steel brine tank containing coils for the passage of the air, or it may be a double-walled casting containing cells for the ice-making cans. These cells are supplied with brine to fill the space between the casting and can, thus conducting heat from the can to the air at a higher rate than that at which it would otherwise pass through an air space. The hollow part of the casting is that through which the air passes. The air is next conducted to the **refrigerating room**, where the temperature need not be so low as that required to make ice. If a low temperature is desired in a room, some of the low-temperature air will have to be taken directly to that room. The air is distributed through the room in coils of pipe *N* and then is taken to a **water butt,** where it cools drinking water to 40° F. or 50° F. The air is then returned to the compressor through the pipe *E*. If the air is still at a low temperature, it is sometimes passed around the pipe *C*, and this cools the air going to the expander so much that a very low temperature is obtained. The pump *O* is the **air-charging pump** driven from the cross-head of the compressor. This pump draws air through

its plunger, and after compression the air is delivered to the trap P, which is surrounded by cold water. This cools the air and causes a large part of the moisture brought in from the atmosphere to be separated and drained off. This air is then delivered to the pipe E and is mixed with the air going to the compressor. The valves Q and R cut off the low-temperature parts when it is desired to operate the by-pass S. The valve T allows hot air to enter the expander from the compressor and thus pass into the trap L, removing grease and snow from it. The trap or separator L has a double bottom or steam jacket which may be used to melt any congealed oil or water, and so open up the line if closed.

To start this machine, the blow-valve on the expander and petcocks on the traps are kept open until no more grease passes through. Then the valves Q and R are opened and S closed. After this T is closed. The circulating water is then turned on, and gradually the low-pressure side should be charged by O until a pressure of 60 lbs. is reached. The high-pressure will then be 210 lbs. The petcock on the water trap P should be opened to keep the water level below the half-full point. The stuffing-boxes, which contain three or four metallic rings, an oiling ring and three or four rings of soft packing, should be supplied with oil. This oil keeps the packing tight with little tightening of the gland, and consequently little friction. These stuffing-boxes are placed at places where the greatest loss of air occurs. The sight-feed lubricators are connected to the stuffing-boxes. The air pistons are packed with cup leathers which last about two months for steady work. They are made of $\frac{1}{8}$ in. thickness and are kept flexible by soaking in castor oil. Once or twice a day the machine is cleaned of oil and grease by opening S and closing Q and R, and then opening T, T' and T''. After this, steam is passed into the jacket of L and the petcock is opened. A blow-off from the expander is also opened. This is done for about one-half hour. A change in the ratio of the two pressures is due to leaky pistons, while a drop in the low pressure is due to leaky stuffing-boxes. These machines are made in small sizes;

the largest are of 3 to 4 tons capacity. They are used chiefly for marine work. There are a number of foreign air refrigerating machines. Such firms as Haslam & Co., J. & E. Hall, and I. & W. Cole are engaged in making these. They vary only in the matter of details from the machine just outlined, and for that reason these will not be described.

The system of refrigeration using a **volatile liquid** is shown in Fig. 13. To be a little more definite, assume that **anhydrous ammonia** is used as the working substance. The **compressor** *A* relieves the pressure in the **coil** *B* by drawing vapor from

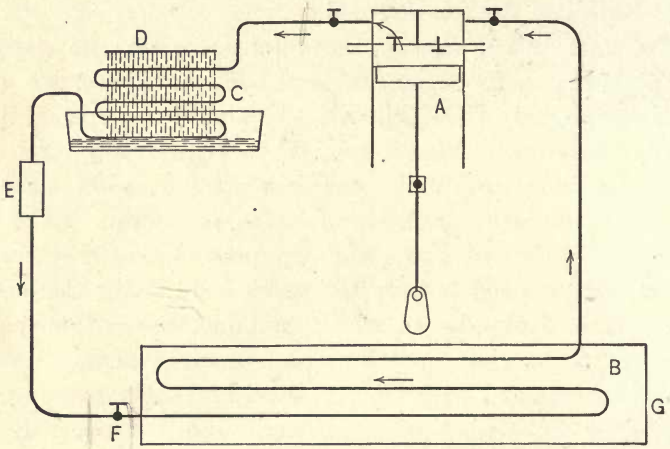


FIG. 13.—Compression Refrigerating Machine.

the coil, since the coil is connected to the suction side of the compressor. The vapor thus removed is compressed by *A* and delivered under pressure into a **condenser** coil *C*. The vapor will be compressed in *C* until the pressure is such that the temperature of saturated ammonia vapor at that pressure is slightly higher (about 10° to 20°) than that of the water coming from the supply *D*. When this pressure is reached, the water, having a lower temperature than that of saturation of the ammonia, will abstract heat from the ammonia and cause it to condense so that liquid ammonia will flow into the **receiver** *E*. If now the pressure in the coil *B* is such that the tempera-

ture of boiling for ammonia at that pressure is above the temperature of the substance around the coil *B*, the ammonia gas and liquid in the coil will give up heat to the substance through the coil and will be cooled off, but nothing further can happen. If, however, the boiling temperature corresponding to the pressure is less than that of the substances around the coil *B*, then heat will flow from them into the ammonia in the coil and cause the liquid to evaporate, requiring the further action of the compressor to keep the pressure low enough to remove heat from the substances near the pipe. Of course, if the first condition were true, no vapor would form and the action of the compressor would reduce the pressure so that the boiling temperature would at least be low enough to remove heat from the substances. The supply of liquid ammonia is regulated by the valve *F*, known as an **expansion valve.** It is in reality a **throttle valve,** throttling the liquid ammonia from the high pressure in *C* to the low pressure in *B*. The coil *B* may be placed in a room to be refrigerated or it may be placed in a tank *G*, containing brine of a low freezing-point. This brine is cooled and sent out to rooms which are to be refrigerated, or to ice tanks, and after receiving the heat, the warm brine is returned to the tank to be cooled again. The first system is known as the **direct-expansion system,** while the latter is called the **brine system of refrigeration.**

The various types of compressors used will be described in a later chapter. At this point, however, one form of compressor will be shown in Fig. 14. This is a steam-driven compressor of the Frick Co.

A **Corliss steam cylinder** *A* drives a shaft *B* with two **cranks.** To the steam-engine crank is connected the rod of one ammonia compressor, while on the other crank at 180° is attached the connecting rod of the other compressor. Thus one steam piston operates two ammonia pistons. In some cases the crank to which the two connecting rods are attached is of the center type; three bearings are then used. This **compressor is single-acting.** Low-pressure ammonia enters at *C* and passes into the cylinder on the up-stroke of the **piston**

D. The long **stuffing-box** *E* is quite a common feature of all compressors, as is the long **piston** with a number of piston rings. These are necessary to prevent the escape of ammonia,

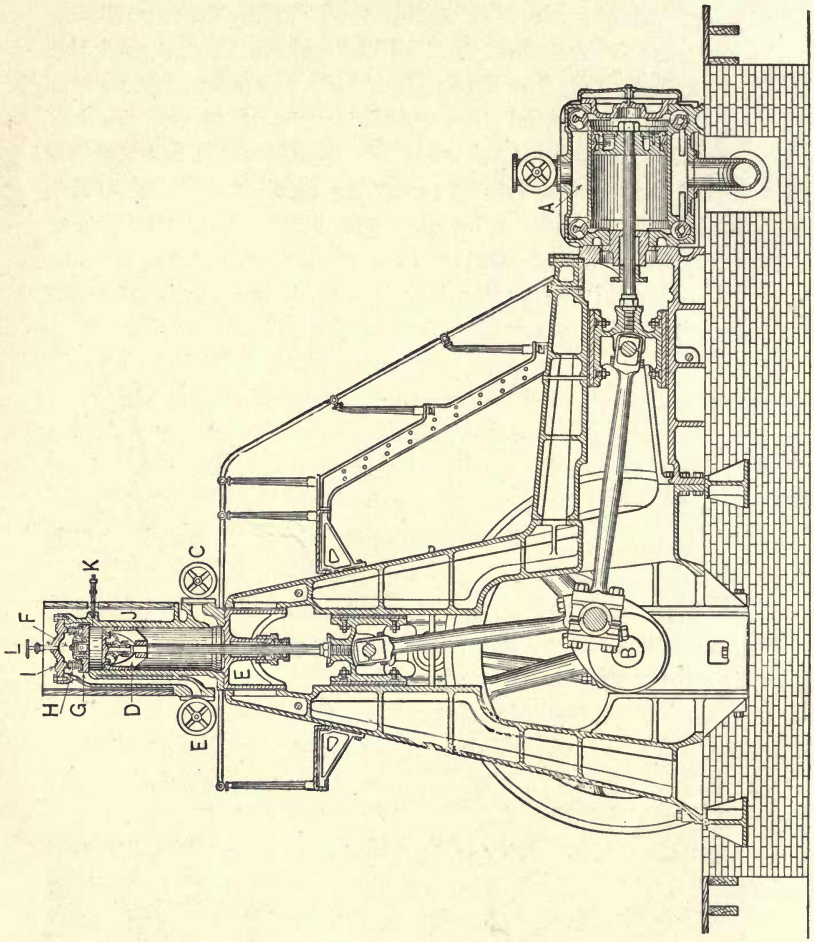


FIG. 14.—Compressor of Frick Co.

which is poisonous and expensive. On the down-stroke of the piston, the large **suction valve** in the center of the piston is opened by the vacuum produced on the upper side of the piston, and vapor is drawn over to that side. On the return stroke of the piston, the vapor is compressed in the cylinder until the

pressure is slightly above that in the **discharge space** *E*, when it opens the **valve** *F* at the center of the head of the cylinder. This valve is forced down by a small spring so that there is only a slight increase in pressure above the line pressure before the valve opens. The vapor pressure in the discharge holds it to its seat on the down-stroke of the piston. The **cylinder head** *G* is not bolted fast to the cylinder barrel. It is held down by the springs *H* which press against the **main head** *I*, attached to the barrel. The purpose of the **safety compressor head** is to avoid the danger of blowing off a head if anything should lodge on top of the piston. If the suction or discharge valve should break, or if scale should accumulate and lie on top of the piston, the small clearance which exists in this compressor would not be large enough to care for this material, and with a rigid head the cylinder would break. With the safety head the springs would yield and permit the head to lift. If for any reason liquid ammonia were to collect and the discharge valve would not relieve it, then the head would rise. Around the cylinder is a **water jacket** *J* for the removal of some of the heat of compression. The **value of the jacket** is questioned by some. If too much water is not used, the heat removed will not have to be taken out in the condenser, and so nothing is lost. Moreover, any heat removed during compression decreases the work, so that there is some saving by the judicious use of the jacket. If much water is used, there will be a loss due to the cost of water being greater than the saving due to the jacket. *K* is an **indicator valve** and *L* is a **purge valve** used to manipulate the compressor.

The layout of a plant using the De La Vergne apparatus is shown in Fig. 15. In this **the vapor,** or, as it is usually called, **the gas,** enters from the **refrigerating rooms** or **brine tank,** and passes to the **suction side** of the compressor. This line is connected to the **gauge board,** where a suction gauge is installed. This gauge is, of course, controlled by a valve to permit of its removal and repair. The suction is sometimes connected to the liquid line by an **equalizing pipe** for manipulation of the plant. The compressed gas then passes over

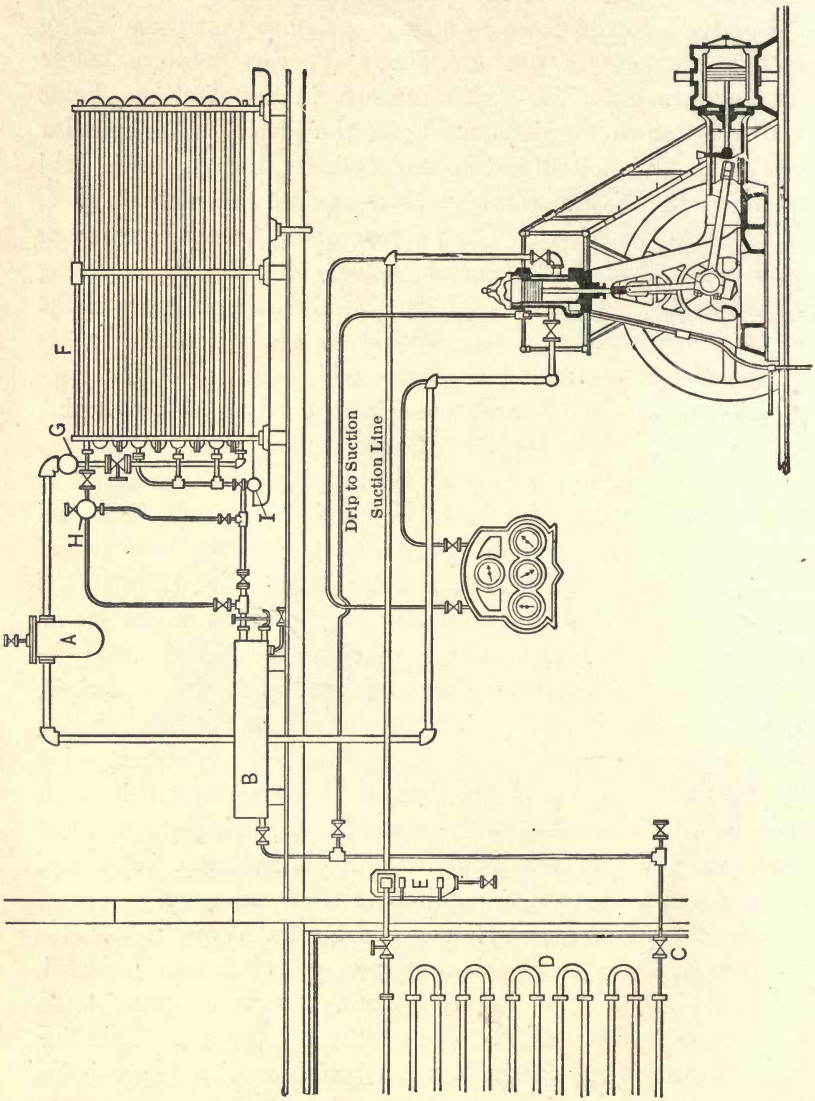


FIG. 15.—Arrangement of De La Vergne Refrigerating System.

to the **condenser.** This is a coil of pipe made of **return bends.** The hot gas enters at the bottom and as it passes upward it is condensed, special bends being used to remove the liquid at different places. These various drip lines are connected, and finally the liquid is delivered to the **storage tank** *B*, from which it discharges through the **expansion valve** *C* into the **expansion coil** *D*. The various euqalizing pipes serve to equalize pressures at various points of the system, so that syphonic action may not be set up. The liquid after entering the refrigerator is changed into vapor and returned to the compressor. The passage through the **scale separator** *E* removes the danger of scoring the cylinders. The valves at the top of the **oil separator** *A* and condenser are to rid the system of **non-condensible gases** which collect there. These gases are due to air which may be drawn in, or from the oil which may be decomposed. In some cases ammonia may be decomposed. The cooling water is discharged from the pipe *F*.

There are other substances used in compression systems. **Sulphur dioxide, carbon dioxide,** and **methyl chloride** are the common ones spoken of to-day. Various ethers and alcohols have been used and certain mixtures of liquids, such as CO_2 and SO_2, have been tried. On account of cost, danger from use, pressures demanded, and sizes of parts, some prefer one substance and some another. The theory underlying all of these is the same, and the description given above would apply to any of them. **In all machines the vapor is raised by compression to such a pressure that the water supply can remove heat from the vapor and condense it, and by use of the throttle valve it is reduced to such a pressure that it will remove heat from a place of low temperature.**

The **absorption machine** depends for its action on the fact that for every concentration of **aqua ammonia,** or for every per cent of a solution of ammonia and water, which is anhydrous ammonia, there exists a certain temperature at which the solution will boil under a given pressure. Thus, if 35% by weight of a solution of aqua ammonia is NH_3, this will boil at 227° F. under a pressure of 170 lbs. gauge, and the

same solution will boil at 110° F. if under 15 lbs. pressure. Now ammonia under 170 lbs. pressure would boil at 91° F., and at 15 lbs. gauge pressure it would boil at 0° F. If water were available at 80° F. and steam at 235° F. or 10 lbs. gauge, the following might be done:

If the aqua ammonia or liquor of 35% **concentration** in the **generator** *A* be heated by the steam at 235° F. in the **steam coil** *B*, the solution will boil and the ammonia and water vapor will produce a gauge pressure of 170 lbs., and this is sufficient to have the ammonia condense in the **condenser** *C* if water at 80° F. is passed over the pipes. If the ammonia collected

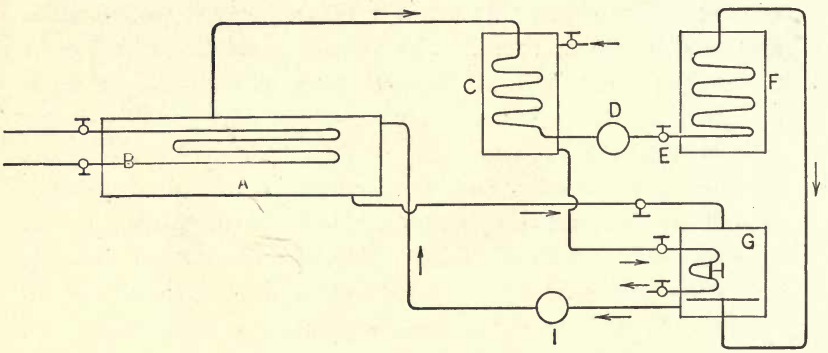


FIG. 16.—Elementary Absorption Machine.

in the **receiver** *D* is passed through the **throttle valve** *E* into the **coil** *F*, where it may abstract heat from brine, it will boil at 0° F., if the pressure is maintained at 15 lbs. If an aqua ammonia solution in the tank *G*, called an **absorber,** is not allowed to get above 110° F. by the **cool water coil** *H*, and is not allowed to get stronger than 35% concentration, it will absorb ammonia and keep the pressure in the absorber and the line leading to *F* at or below 15 lbs. gauge. To keep the solution in the absorber in condition to absorb ammonia, the **weak liquor** in the generator, from which the ammonia was removed, is allowed to flow into the absorber, the **pump** *I* forcing the strong liquor from *G* to *A*.

This is the explanation of the simple absorption machine, but there are several refinements which are used. Certain phenomena will have to be described. When an aqua solution boils, not only ammonia escapes, but also water vapor. Moreover, the heat supplied will have to be not only that required to drive off the ammonia (heat of solution) and that required to evaporate the moisture, but also enough to superheat the ammonia vapor and water vapor, since these leave the generator in a superheated condition. This excess of superheat must be removed and to reduce the amount of heat to be taken out by cooling water, and to reduce the heat supply to the generator, the cool strong solution coming from the absorber is caused to flow over trays through which the heated gases pass from the generator *A*. In this way the liquor is heated and economy effected. This apparatus is known as the **analyzer, *K***.

The water vapor condensing in the condenser would absorb some ammonia and reduce the efficiency of the apparatus. To reduce this loss it is customary to pass the vapors leaving the analyzer through tubes *L* over which the cool, weak solution from the absorber, or water from the condenser, flows. In this way the temperature of the mixture of ammonia and water vapor is so reduced that most of the water vapor is condensed and separated by the **separator *M***, and sent back to the analyzer. *L* is known as a **rectifier** or **dehydrator.** Of course, this water absorbs ammonia and reduces the amount sent to the condenser, but it is not delivered to the condenser and so causes no trouble. The last change which is introduced for economy is to pass the warm, weak solution, which is to go to the cooled absorber, around pipes carrying from the absorber the cool, strong liquor, which has to be heated. This interchange saves much heat. The apparatus is known as the **interchanger, *J***.

Before passing to the actual arrangement, one other point must be mentioned. The solution is changed from one concentration to another in the absorber and in the generator, and it must be remembered that it is the weak concentration which

fixes conditions in the generator and the strong concentration, those in the absorber.

The **heat of solution** when aqua ammonia changes from one concentration to another must be cared for by the cooling coil in the absorber, so that there is no increase in temperature, which would cut down the possible concentration. The computations for this system will be given in the next chapter.

Fig. 17 shows the absorption machine diagrammatically. In this arrangement the strong solution in *A* is boiled by the steam in *B*. The vapor passes through the **analyzer** *K*, where

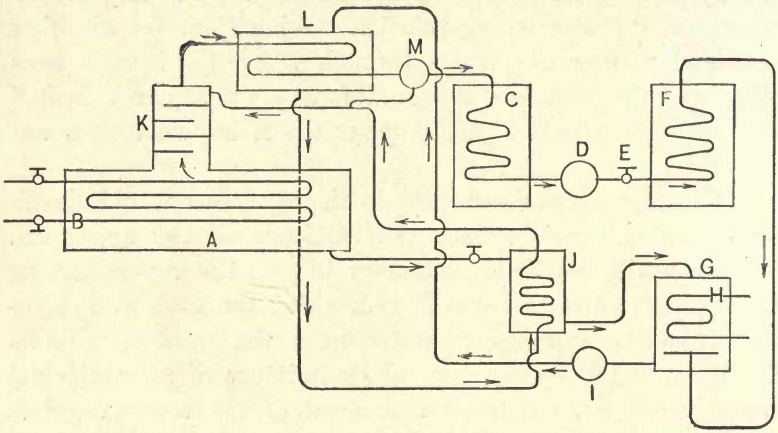


FIG. 17.—Complete Diagram of the Absorption Machine.

it meets the down current of warmed strong liquor coming from the **exchanger** *J*. This cools the vapor and warms the liquor, and, it may be, drives off some ammonia. The vapor then passes to the **rectifier** or **dehydrator,** *L*, which is cooled by the strong liquor pumped by pump *I* from the absorber. This solution is cool enough to condense most of the moisture in the vapors. The water formed absorbs ammonia, and this liquor is removed by the **separator** *M*, and is passed back to the **analyzer.** The liquid from the condenser *C* passes to the **receiver** *D* and through the **valve** *E* to the **expansion coil** *F*, in the **brine tank.** From the absorber *G* with its cooling coil *H* the liquor is pumped by *I* to *L* and then to the **inter-**

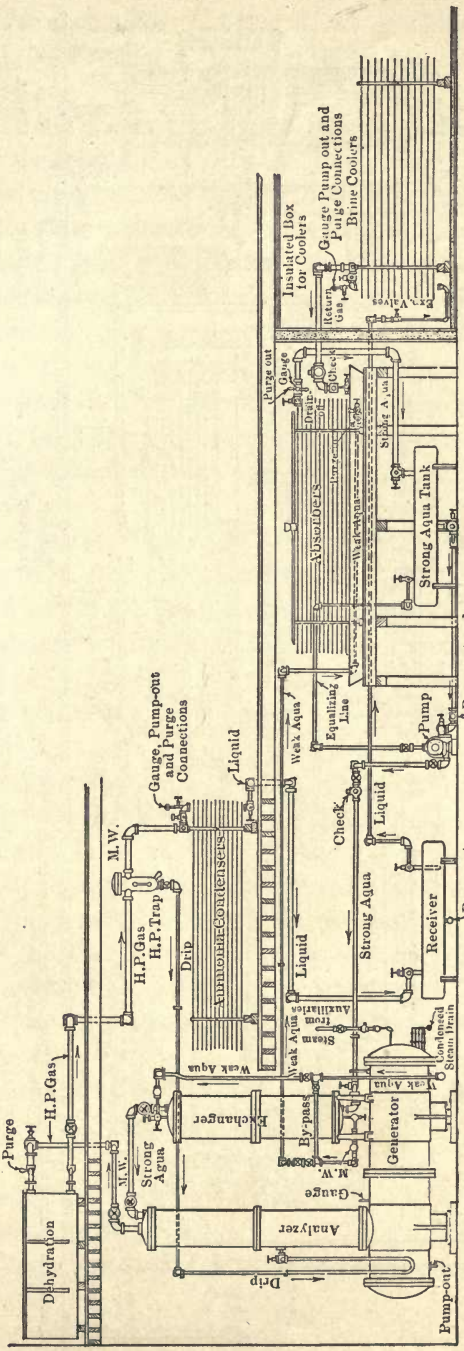


FIG. 18.—York Absorption Apparatus.

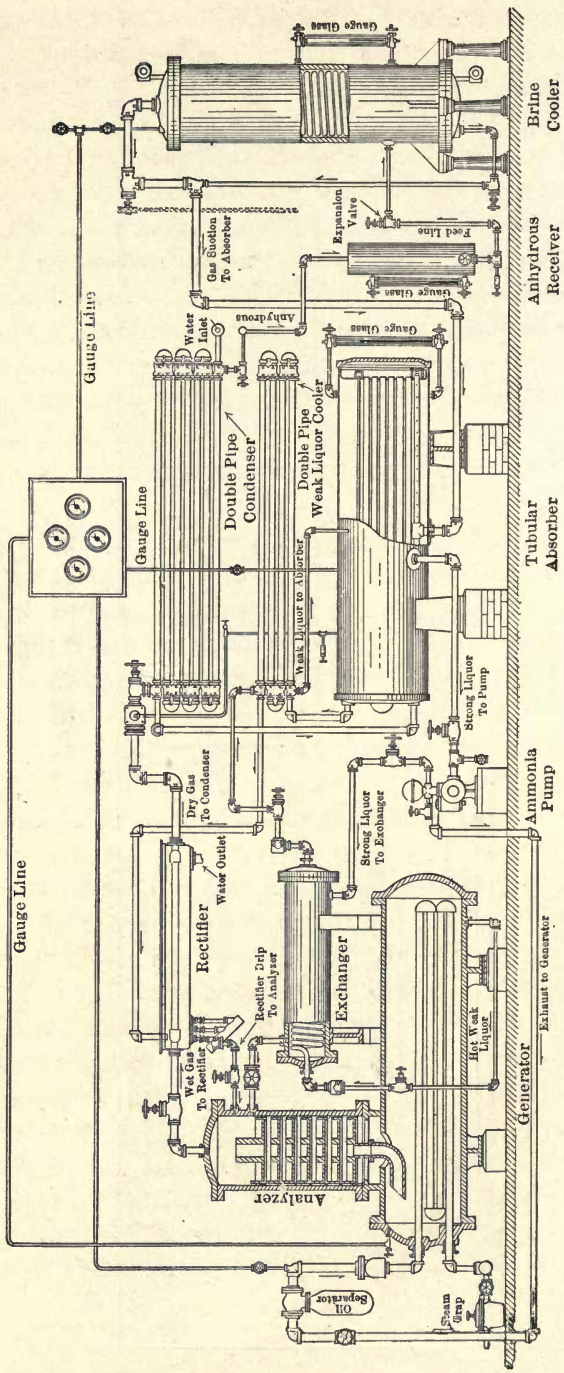


FIG. 19.—Absorption Refrigerating Machine, Double Pipe Type. Carbondale Machine Co.

changer or **exchanger** *J*, after which it enters the analyzer. The weak liquor from *A* passes through the **exchanger** *J* to the **absorber** *G*.

Fig. 18 illustrates the construction of an actual absorption machine made by the York Manufacturing Company. The condenser, absorber, and expansion coils are all of the exposed-coil type. The water lines going to condenser, dehydrator, and absorber are not shown. **Purging valves** are shown at various high points. They are connected by purge lines which are not shown. For purging this system the best location is at the absorber, for here all of the ammonia is absorbed, and the gas remaining is true **non-active gas.** The various parts of the apparatus, especially the expansion coils, should be purged into the absorber, so that any ammonia coming over may be condensed.

Fig. 19 shows the equipment of an absorption plant as made by the Carbondale Machine Co. This differs from that of Fig. 18 in that a **double-pipe condenser** is used, instead of an atmospheric one; the absorber is a **tubular absorber;** the weak liquor is cooled before entering the absorber and the expansion coil is in a **brine cooler.** The action of the apparatus can be followed from the description above.

The **evaporation** of a portion of a body of water, so as **to freeze** the remaining portion, has been used in a natural way for centuries. One of the first successful machines acting on this principle was made by Edmund Carré about the middle of the last century. The apparatus consisted of an air pump attached to a cylindrical vessel containing sulphuric acid. A carafe containing water was attached to the vessel by a hose and then the air pump was started. As the pressure was reduced, the water in the carafe would boil, due to its own heat, and the sulphuric acid would absorb the water vapor, lowering the pressure still further. The heat of vaporization of the vaporized water would be taken from the water in the carafe until finally this removal of heat would cause the remaining water to be frozen.

This invention has been followed by a number of patents,

and some actual installations. John Patten has invented apparatus for ice manufacture on a commercial scale, but the value of such installations has not been proven. A. J. Stahl has used a Patten plant at South Bend, Indiana, for the production of 30 tons a day. The drawback is to obtain the high vacuum necessary for freezing. Water boils at 32° F. when the pressure is less than 0.08 lb., or 0.16 in. of mercury. A still lower pressure is required to freeze ice in a short time. That such ice is pure is assumed, because the expansion of the gases within living organisms at this low pressure causes the organism to rupture. Under very low pressures the evaporation is so rapid that the ice forms immediately.

In one type of machine, patented by J. H. J. Haines in 1901, an air pump was attached to a vessel containing water to be frozen, to the space outside this vessel and within an iron receptacle, and to a vessel containing sulphuric acid. As the air pump was operated, the whole system was exhausted. The space outside the water vessel being exhausted, prevented heat from passing across and so, when the pressure was low enough for the water to boil by its own heat, this heat could come only from the water, since the vessel was well insulated from the outside by the vacuum space. This abstraction of heat caused the remaining water to freeze. The acid in the third vessel absorbed the water vapor formed and thus reduced the pressure.

In the apparatus patented by W. T. Hoofnagle in 1903, water to be made into ice passes through a vessel *A*, Fig. 20, which may be used to clear the water or filter it. It enters the chamber *B*, in which it is sprayed over a series of trays; the fan at the bottom keeps up a circulation. The chamber is connected to the intermediate cylinder *D* of a three-stage compressor or air pump. This reduces the pressure in *B* to such an extent that the water is de-aerated, and a small amount of evaporation cools the water. This is admitted from the pipe *E* into a chamber *F* by two valves. The chamber *F* is connected to the low-pressure cylinder *G* of the three-stage vacuum pump or compressor. The air and water vapor drawn

out by *G* is sent to *D* and after compression it is finally delivered by *H*. In the chamber *F* are two trays which are oscillated by rods operated by cams. When in the position shown, water is discharged from the nozzles on these trays, and as it flows along the tray the evaporation of water under the very low pressure causes the remaining water to freeze, making a cake of ice.

It is considered advisable in all these machines to spread the water in a thin layer so that freezing may take place readily,

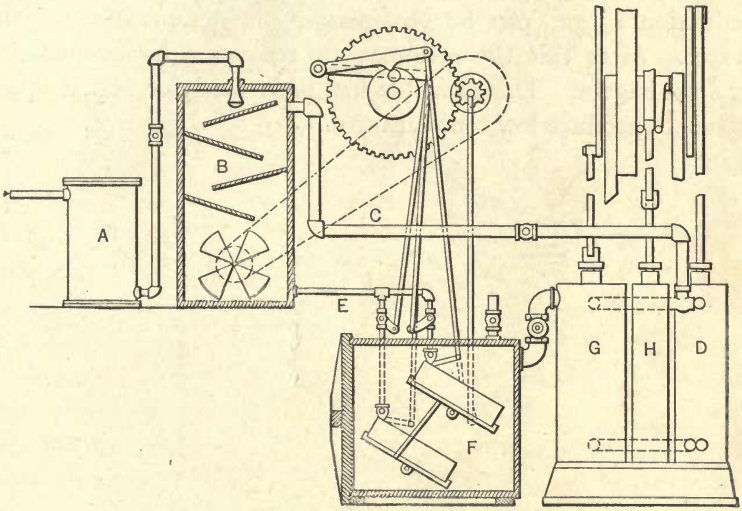


FIG. 20.—Hoofnagle Vacuum System of Ice Making.

as evaporation can occur over a large surface. Patten sprays his water from a movable head.

Another machine for which claims are made is that due to Le Blanc. In this the vacuum is obtained by steam aspirators in series, each compressing the air exhausted by the previous one. By using the Le Blanc condenser pump, this apparatus is used to advantage. In this country the development of this machine is being made by the Westinghouse-Le Blanc interests. This apparatus is being developed for use on ships of the French navy, on some of which serious accidents occurred by the

bursting of the parts of other forms of refrigerating apparatus. The description of these machines is found in *Ice and Refrigeration*, for July, 1912, and Aug., 1910, and *Power*, Jan. 11, 1916.

The **chemical process** refers to the cooling of water by the addition of some soluble chemical. If ammonium nitrate is added to water, the temperature of the solution is much lower than that of the water, a temperature decrease of 40° being obtainable in this way. If the vessel containing the solution surrounds some object, heat will be abstracted and even ice may be formed. If calcium chloride is dissolved in water, a reduction of 30° can be obtained in the temperature of the water. After this the salt may be recovered by evaporation and used again. This principle has been used for actual apparatus to produce low temperatures.

CHAPTER III

THERMODYNAMICS OF REFRIGERATING APPARATUS

WHEN ice is made it is difficult to tell at just what temperature some of the ice forms, and also, after forming, it may be possible to reduce the temperature of some of the ice. Hence

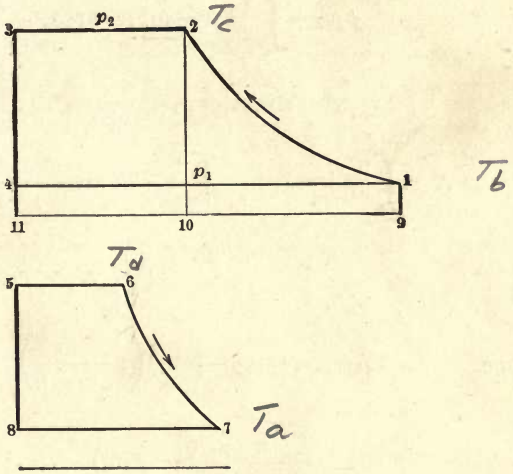


FIG. 21.—Cards from Air Machine.

the amount of heat required to form a pound of ice is not definite. However, if ice melts, the melting does not begin until 32° is reached, and it continues at this temperature until all of the ice is melted. For these reasons the amount of heat required to melt a pound of ice is used as a unit rather than that removed to make a pound of ice.

Refrigeration is usually measured in **tons of ice-melting capacity per twenty-four hours.** Since the **latent heat of fusion of ice** is 143.4 B.t.u. per pound, according to the latest

experiments, this unit means the removal of 286,800 B.t.u. per twenty-four hours, or 199.2 B.t.u. per minute.

The air refrigerating machine has a compressor and an expander, the indicator cards of which are shown in Fig. 21. The expansion and compression curves are of the form $pv^n =$ const., and the compression is from the pressures p_1 to p_2. The cards are shown with no clearance. The area of the card is

Area 3–2–10–11 + area 10–2–1–9 − area 4–1–9–11

or

$$p_2v_2 + \int_{v_2}^{v_1} pdv - p_1v_1 = \text{area}, \quad \ldots\ldots (1)$$

$$pv^n = \text{const.} = p_1v_1^n = p_2v_2^n. \quad \ldots\ldots (2)$$

$$\therefore \int_{v_2}^{v_1} pdv = \text{const.} \int_{v_2}^{v_1} v^{-n}dv = \frac{\text{const. } v^{1-n}}{1-n}$$

$$= \frac{p_1v_1 - p_2v_2}{1-n}. \quad \ldots\ldots\ldots (3)$$

Hence $$\text{Area} = (p_2v_2 - p_1v_1)\left(1 - \frac{1}{1-n}\right) \quad \ldots\ldots (4)$$

$$= \frac{n}{n-1}(p_2v_2 - p_1v_1) \quad \ldots\ldots\ldots (5)$$

$$= \frac{n}{n-1}MB[T_2 - T_1], \quad \ldots\ldots\ldots (6)$$

since $pv = MBT$ for perfect gases,

where p = pressure in lbs. per sq.ft.;
v = volume in cu.ft.;
M = lbs. of gas;
B = constant $= \frac{1544}{\text{mol. wt. gas}}$;
T = absolute temperature in deg. F.

By the thermodynamic theory $B=\frac{k-1}{k}Jc_p$.

Hence

$$\text{Area}=\text{work}=\frac{k-1}{k}\,\frac{n}{n-1}MJc_p[T_2-T_1]. \quad . \quad . \quad . \quad (7)$$

The **temperature** T_1 is fixed by the temperature desired in the refrigerator. If the system is open (air discharged from expander into cold room), T_1 is the temperature of the room, while if a closed system is used, T_1 must be about 10° below the cold room, or the warmest place refrigerated, since

$$p_2v_2^n=p_1v_1^n,$$

$$T_2p_2^{\frac{1-n}{n}}=T_1p_1^{\frac{1-n}{n}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

or

$$T_2=T_1\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

The work of the expansion cylinder is

$$\text{Work}=\frac{k-1}{k}\,\frac{n}{n-1}MJc_p[T_6-T_7]. \quad . \quad . \quad . \quad (10)$$

In this T_6 is fixed by the cooling water being about 10° F. above the water on the opposite side of the cooler metal at the point where the air leaves.

In most compressors and expanders the action is so rapid that the compression is adiabatic and n equals k. This gives

$$\text{Work}_{\text{comp}}=MJc_p(T_2-T_1), \quad . \quad . \quad . \quad . \quad . \quad (11)$$

$$\text{Work}_{\text{exp}}=MJc_p(T_6-T_7). \quad . \quad . \quad . \quad . \quad . \quad (12)$$

If there is friction of 100 f per cent, the net work required to drive the machine is

$$\text{Net work}=MJc_p\left[\left(\frac{1}{1-f}\right)(T_2-T_1)-(1-f)(T_6-T_7)\right]. \quad (13)$$

With no friction the work is

$$\text{Net work} = MJc_p[(T_2 - T_1) - (T_6 - T_7)], \quad . \quad . \quad (14)$$

$$T_7 = T_6\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

The expander cylinder should always be carefully lagged to prevent the entrance of heat, but in the compressor the use of the water jacket reduces the compression line by abstracting heat and thus saving work. If it is assumed that the exponent

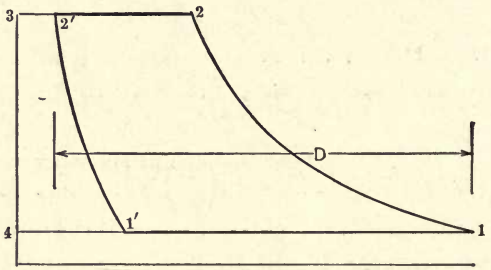


FIG. 22.—Card with Clearance.

$n = 1.35$ in the compressor, while $k = 1.4$, the expression for net work with friction becomes

Net work

$$= MJc_p\left[\left(\frac{1.35}{.35}\frac{.4}{1.4}\right)\left(\frac{1}{1-f}\right)(T_2 - T_1) - (1-f)(T_6 - T_7)\right] \quad (16)$$

$$= MJc_p\left[(1.10)\left(\frac{1}{1-f}\right)(T_2 - T_1) - (1-f)(T_6 - T_7)\right], \quad . \quad (17)$$

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{.35}{1.35}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

$$T_7 = T_6\left(\frac{p_1}{p_2}\right)^{\frac{.4}{1.4}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

The **effect of clearance** is seen from Fig. 22, in which the compression 1—2 is followed by the discharge 2—2′, and then

the air $2'3$, which is retained in the clearance space, expands from $2'$ to $1'$. The net work is $1\text{–}2\text{–}2'\text{–}1'$, and the net amount of air drawn in is $1'\text{–}1$. The temperature at $2'$ is that at 2, hence that at $1'$ is the same as at 1.

$$\text{Work} = M_1 \frac{k-1}{k} \frac{n}{n-1} Jc_p(T_2 - T_1) - M'_1 \frac{k-1}{k} \frac{n}{n-1} Jc_p(T'_2 - T'_1)$$

$$= (M_1 - M'_1) \frac{k-1}{k} \frac{n}{n-1} Jc_p(T_2 - T_1), \quad \ldots\ldots \quad (20)$$

$$= M \frac{k-1}{k} \frac{n}{n-1} Jc_p(T_2 - T_1), \ldots\ldots\ldots \quad (21)$$

where $M = M_1 - M'_1$, or the weight taken in from 1 to $1'$.

This expression is the same as that in Eq. (7) without clearance. There is no effect of clearance on the work of a compressor for which the expansion and compression lines are complete, and have the same exponent. This may be said of an expander also, when the cutoff is such that the expansion is complete or just reaches the back pressure at the end of the stroke, and the point of compression is such that the compression is just carried to the initial pressure.

The above is true as far as indicated work is concerned, but the work required to drive the compressor is slightly greater with clearance, as displacement must be increased for a given discharge if clearance is present, and there is consequently more friction. Let the clearance $2'-3$ be l times the displacement $V_1 - V'_2$ or D.

That is, let

$$3 - 2' = lD.$$

The volume of air taken in is $V_1 - V'_1$, or

$$V_1 - V'_1 = V = D + lD - lD\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}},$$

$$= D\left[1 + l - l\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\right]. \quad \ldots\ldots \quad (22)$$

The expression within the bracket is known as the **clearance factor.**

$$D_{comp} = \frac{V}{1+l-l\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}} = \frac{MBT_1}{p_1\left(1+l-l\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\right)} \quad . \quad . \quad . \quad (23)$$

For the expander with complete expansion and compression

$$D_{exp} = \frac{MBT_7}{p_1\left(1+l-l\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

The refrigeration is produced by adding heat to the air, increasing the temperature from T_7 to the original T_1, at constant pressure. Hence

$$\text{Refrigeration} = Mc_p(T_1 - T_7) = 199.2 \times \text{tons of ref.}; \quad . \quad . \quad . \quad (25)$$

M = weight of air required per minute;

c_p = specific heat at constant pressure = 0.24 for air.

The heat removed in the cooler is given by

$$\text{Cooling} = Mc_p(T_2 - T_6) = G(q'_{out} - q'_{in}). \quad . \quad . \quad (26)$$

M = weight of air per minute;

G = weight of cooling water per minute;

q'_{out} = heat of liquid of cooling water at outlet temperature;

q'_{in} = heat of liquid of cooling water at inlet temperature.

The expression, in B.t.u.'s, $\frac{\text{Refrigeration}}{\text{Work}}$ is known as the **refrigerating effect.** With no friction this becomes

$$\frac{Mc_p(T_1 - T_7)}{Mc_p[(T_2 - T_1) - (T_6 - T_7)]} = \frac{1}{\frac{T_2 - T_6}{T_1 - T_7} - 1} \quad . \quad . \quad . \quad (27)$$

Now

$$\frac{T_2 - T_6}{T_2} = \frac{T_1 - T_7}{T_1},$$

since

$$\frac{T_2}{T_1}=\frac{T_6}{T_7}=\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}.$$

$$\therefore \quad \text{Ref. eff.}=\frac{1}{\frac{T_2}{T_1}-1}=\frac{T_1}{T_2-T_1}.$$

This shows that as T_2-T_1 becomes smaller the refrigerating effect or the refrigeration per unit of work increases.

The general expression is

$$\text{Ref. eff.}=\frac{T_1-T_7}{\left[(1.10)(T_2-T_1)\left(\frac{1}{1-f}\right)-(T_6-T_7)(1-f)\right]}. \qquad (28)$$

If a problem is given the following steps are taken: (*a*) T_1 is fixed by the temperature of the room or place to be cooled and T_6 by the temperature of the cooling water. Of course the coldest water should be brought in contact with the coldest air, or the air and water must flow in opposite directions along the cooling surface, giving a **counter-current flow.** (*b*) By (18) and (19) the temperatures T_2 and T_7 are found after assuming the ratio $\frac{p_2}{p_1}$. It will be noted that T_2 and T_7 depend upon the ratio $\frac{p_2}{p_1}$, and not upon the actual values of the pressures. (*c*) By (25) M, the weight of air per minute, is found for a given number of tons of refrigeration. (*d*) G, the amount of cooling water per minute, is found by (26). (*e*) The displacement of the compressor per minute is found by (23). (*f*) The displacement of the expander per minute is given by (24) and the horse-power is given by dividing (17) by 33,000.

H.P. to drive machine =

$$\frac{MJc_p}{33{,}000}\left[1.10\left(\frac{1}{1-f}\right)(T_2-T_1)-(1-f)(T_6-T_7)\right]. \qquad (29)$$

There are some changes to be noticed before proceeding with a problem. If in the expander the cutoff is too late,

the areas 7–10–9 and 11–12–13 will be lost, decreasing the work done by the expander and increasing the net work done by motor which drives the machine. Moreover, T_7 is now higher than it was when the air was expanded down to the lowest pressure. This gives less refrigeration, since the free expansion 7 to 9 is throttling action and will not cool the air. The temperature at 13 is higher than the temperature T_6 of the incoming air, because T'_7 is higher than T_7 would have been if the expansion were complete. This even makes T_6

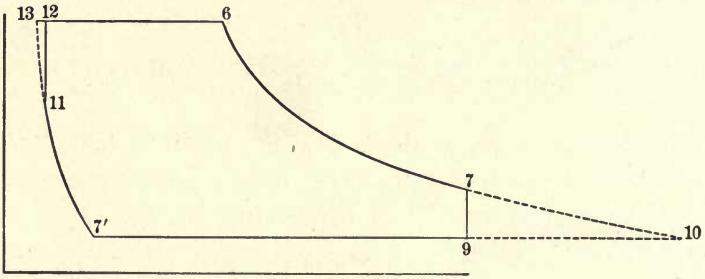


FIG. 23.—Incomplete Expansion and Compression in Expander.

higher than it should have been, increasing T_7 and still further cutting down the refrigeration

$$Mc_p(T_1 - T_7).$$

This incomplete action makes the displacement $V_9 - V_{12}$ less than it would be with complete expansion and compression, $V_{10} - V_{13}$. This is the only advantage.

Since one would be foolish to design an expander with incomplete expansion and compression, this will not be further investigated although by calculations the various temperatures may be found for any conditions.

Note.—The following discussion gives the temperatures, assuming no compression, with expansion such that $p_7 = 2p_1$.

Air entering at temperature T_6 must compress the clearance air lD, at temperature $T_{7'}$, from p_1 to p_2.

The energy in the air contained in the cylinder is $\frac{p_1 lD}{k-1}$.

The air brought in from the storage tank to bring this up to the pressure p_2 is m lbs., and the energy entering from the tank at constant pressure is mc_pT_6. The energy after mixing is $\frac{p_2lD}{k-1}$.

Hence
$$Jmc_pT_6+p_1\frac{lD}{k-1}=\frac{p_2lD}{k-1}.$$

$$m=\frac{(p_2-p_1)\frac{lD}{k-1}}{Jc_pT_6}. \quad \ldots\ldots\ldots \quad (30)$$

Now the weight of air in the clearance space is

$$m'=\frac{p_1lD}{BT_{7'}}. \quad \ldots\ldots\ldots \quad (31)$$

The temperature of the mixture in the clearance space, after the air enters to fill the space, is given by

$$T_{6'}=\frac{p_2lD}{(m+m')B}=\frac{p_2lD}{\left[\frac{(p_2-p_1)\frac{lD}{k-1}}{Jc_pT_6}+p_1\frac{lD}{BT_{7'}}\right]B}=\frac{p_2}{\frac{(p_2-p_1)B}{J(k-1)c_pT_6}+\frac{p_1}{T_{7'}}}$$

$$=\frac{p_2}{\frac{(p_2-p_1)}{kT_6}+\frac{p_1}{T_{7'}}}=\frac{kp_2T_6T_{7'}}{p_2T_{7'}+p_1(kT_6-T_{7'})}. \quad \ldots\ldots\ldots \quad (32)$$

When the remaining air $(M-m)$ passes into the cylinder, the following equation is true:

$$(M-m)c_pT_6+(m+m')c_vT_{6'}+Ap_2lD=\frac{k}{k-1}Ap_2V_{6''};$$

$$(M-m)c_pT_6+(m+m')c_vT_{6'}+AB(m+m')T_{6'}=\frac{k}{k-1}AB[M+m']T_{6''};$$

$$(M-m)c_pT_6+(m+m')c_pT_{6'}=(M+m')c_pT_{6''}; \quad \ldots\ldots\ldots \quad (33)$$

$$T_{6''}=\frac{(M-m)T_6+(m+m')T_{6'}}{M+m'}. \quad \ldots\ldots \quad (34)$$

In this $T_{6'}$ is higher than T_6 and $T_{6''}$ is higher than T_6. If now the air at volume $V_{6''}$ expands from pressure p_2 to a pressure $2p_1$, the temperature of discharge will be

$$T_7=T_{6''}\left(\frac{2p_1}{p_2}\right)^{\frac{n-1}{n}}.$$

This will be the temperature of discharge, since the free epxansion is a throttling action of constant temperature. Even if $T_{6''}$ were equal to T_6, T_7 would be higher than it should be by the factor $2^{\frac{n-1}{n}}$. This means that the refrigeration is decreased.

The work returned by the expander in this case is

$$J(M+m)c_p(T_6-T_7)+\frac{(2p_1-p_1)(M+m)BT_7}{2p_1}-(p_2-p_1)lD. \quad . \quad (35)$$

These quantities may all be computed.

The compressors, as usually constructed and used, operate in such a manner that complete expansion and compression result, and consequently there is no effect, due to clearance space, on the temperature or work of that part of the apparatus.

The **effect of moisture** in the air is to reduce the refrigerating effect and increase the net work. Of course, this moisture effect is not felt if the same air is used over and over again, since the first chilling dries the air and removes the moisture.

Air contains a certain amount of moisture. The amount is told by a **hygrometer**. One form of this apparatus consists of two thermometers, one of which has a wet wicking around it. If now, these two thermometers are whirled in the air, it will be found, usually, that the **wet-bulb thermometer** will read less than the **dry one.** The amount by which the wet bulb is lowered depends on the moisture present in the air. If the air is saturated with moisture, there will be no difference, while if the air is dry, there is a large difference in the temperatures recorded by each. The amount of moisture is designated by **relative humidity.** Relative humidity is the ratio of the amount of moisture, m_a, in a cubic foot of air compared with the amount of moisture, m_s, to saturate it, or

$$\rho=\frac{m_a}{m_s}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (36)$$

ρ = relative humidity;

m_a = amount of vapor in 1 cu.ft.;

m_s = amount of vapor to saturate 1 cu.ft.

If the **vapor pressure** (**vapor tension** or **steam pressure**) at a given temperature is p_s, the actual pressure, p_a, exerted by the vapor of relative humidity ρ is

$$p_a = \rho p_s.$$

Now, if the wet bulb reads t_w and the dry bulb t_a, and the barometer reading is given by Bar, then according to Carrier

$$\rho = \frac{p_w}{p_a} - \frac{\text{Bar} - p_w}{p_a} \times \frac{t_a - t_w}{2755 - 1.28 t_w}. \quad . \quad . \quad . \quad (38)$$

where ρ = relative humidity;
p_w = steam pressure corresponding to t_w;
p_a = steam pressure corresponding to t_a;
Bar = barometer reading;
t_a = dry-bulb reading;
t_w = wet-bulb reading.

If the air of relative humidity ρ and temperature T_1 enters the compressor, the moisture and air during compression will act as a single gas and the temperature T_2 will be found as before.

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} \quad \text{or} \quad T_1 \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}.$$

The work of compression will be

$$J[Mc_p + mc'_p][T_2 - T_1], \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

$$M = \frac{(p_1 - \rho p_s)V_1}{BT_1}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

$$m = \rho m_s V \quad \text{or} \quad \frac{\rho p_s V_1}{B_s T_1},$$

m_s = weight of 1 cu.ft. of saturated steam at temperature T_1;
p_s = saturation pressure of steam at temperature T_1;

$$B_s = \frac{1544}{18} = 85.8.$$

When this air is cooled to temperature T_2, an investigation must be made to see if any of the moisture is condensed. If none is condensed the mixture acts as a gas and

$$V_6 = V_2\left(\frac{T_6}{T_2}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

If now $V_6 m_{s6} > m$ no moisture is condensed and

$$\rho_6 = \frac{m}{V_6 m_{s6}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

$V_6 m_{s6} < m$ for low pressures, and then $m - m_{s6} V_6 =$ amount condensed in the cooler.

In the first case m lbs. enter the expander and in the second case $m_{s6} V_6$ enters the expander.

This moisture is condensed and frozen as soon as it enters the expander. It then gives up the heat

$$m(q'_6 + r_6 + 143.4) = c, \quad . \quad . \quad . \quad . \quad . \quad (43)$$

to the cylinder walls and leaves the volume of the air

$$V'_6 = \frac{MBT_6}{p_2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

The heat c taken up by the cylinder walls may be assumed to be gradually restored during expansion. If this is divided between the temperatures T_6 and T_7, it might be assumed that the amount returned from the walls per dt degrees is

$$+\frac{q'_6 + r_6 + 143.4}{T_6 - T_7} m dt = +ma dt,$$

where a is the value of the fraction.

Hence the equation for work at the expense of internal energy and the heat returned is

$$Jmc_t dt + JMc_v dt + ma dt = -p dV, \quad . \quad . \quad . \quad (45)$$

$$J[mc_i+Mc_v+ma]\frac{dt}{BT}=-M\frac{dV}{V}$$

$$\frac{Jc_v}{B}\left[1+\frac{m(c_i+a)}{Mc_v}\right]\log_e\frac{T_6}{T_7}=\log_e\frac{V_7}{V_{6'}}, \quad . \quad . \quad . \quad (46)$$

$$\frac{1}{k-1}\left[k+m\frac{(c_i+a)}{Mc_v}\right]\log_e\frac{T_6}{T_7}=\log_e\frac{V_7}{V_{6'}}+\log_e\frac{T_6}{T_7}=\log_e\frac{p_2}{p_1}. \quad (47)$$

From this T_7 may be found.
The work done is

$$\begin{aligned}W_{ex}&=p_2V_{6'}+\int_{V_{6'}}^{V_7}pdv-p_1V_7\\&=MBT_6+J(mc_i+Mc_v+ma)(T_6-T_7)-MBT_7\\&=J[mc_i+M(c_v+AB)+ma][T_6-T_7]\\&=J[mc_i+Mc_p+ma][T_6-T_7]. \quad . \quad . \quad . \quad . \quad . \quad . \quad (48)\end{aligned}$$

The net work is

$$\begin{aligned}\text{Work}_n=J\Bigg[&[Mc_p+mc'_p][T_2-T_1]\left[\frac{1}{1-f}\right]\\&-[Mc_p+mc_i+ma][T_6-T_7][1-f]\Bigg]. \quad . \quad (49)\end{aligned}$$

The refrigeration is

$$JMc_p[T_1-T_7]. \quad . \quad . \quad . \quad . \quad . \quad . \quad (50)$$

The refrigerating effect is

$$\frac{Mc_p[T_1-T_7]}{\text{Work}_n}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (51)$$

From the expression for refrigeration it is noted that only air is considered to be delivered back to the compressor, and consequently unless this air received moisture from the space through which it was passed, this problem is of little value.

If a large quantity of water is injected into the cylinder during compression to reduce the amount of work by cooling, of course the above discussion of the expander would be im-

portant, and of course some means would have to be used to remove the ice formed constantly in the cylinder.

If the water injected into the cylinder at each stroke is m pounds, the following equations should hold for the compression stroke.

$$J[Mc_v dt+md(q'+x\rho)]=-PdV=-(P-p)dV-pdV, \quad (52)$$

$$J[Mc_v dt+mdq'+m\rho dx+mxd\rho]=-pdV-(P-p)dV,$$

$$(P-p)dV=\frac{MBT}{V}dV;$$

$$dq'=dt, \text{ approximately;}$$

$$dV=m[(v''-v')dx+xd(v'')].$$

$$\therefore \quad Jm\rho dx+pdV=Jm[\rho+Ap(v''-v')]dx+mpxdv''$$

$$=Jm(rdx+Apxdv''). \quad (53)$$

Now

$$d\rho=dr-Apdv''-A(v''-v')dp,$$

$$xd\rho=xdr-Apxdv''-Ax(v''-v')dp. \quad (54)$$

But

$$Ax(v''-v')dp=\frac{xr}{T}dt. \quad (55)$$

Hence

$$xd\rho+Apxdv''=xdr-\frac{xrdt}{T}. \quad (56)$$

$$\therefore \quad JM\left[\left(c_v+\frac{m}{M}\right)dt+\frac{m}{M}\left(rdx+xdr-\frac{xrdt}{T}\right)\right]=-\frac{MBT}{V}dV$$

$$JM\left[\left(c_v+\frac{m}{M}\right)\frac{dt}{T}+\frac{m}{M}\left(\frac{r}{T}dx+\frac{xdr}{T}-\frac{xrdt}{T^2}\right)\right]=-MB\frac{dV}{V}. \quad (57)$$

$$[Mc_v+m]\log_e\frac{T_2}{T_1}+m\left[\frac{x_2r_2}{T_2}-\frac{x_1r_1}{T_1}\right]=AMB\log_e\frac{V_1}{V_2}. \quad (57')$$

$$V_1=\frac{MBT_1}{P_1-p}, \quad (58)$$

$$V_2 = \frac{MBT_2}{P_2 - p_2}, \quad \ldots\ldots\ldots \quad (59)$$

$$V_2 = mx_2 v''_2, \quad \ldots\ldots\ldots \quad (60)$$

$$V_1 = mx_1 v''_1. \quad \ldots\ldots\ldots \quad (61)$$

$$[Mc_v + m] \log_e \frac{T_2}{T_1} + \frac{V_2 r_2}{v_1'' T_2} - \frac{V_1 r_1}{v_1'' T_1} = AMB \log_e \frac{V_1}{V_2}. \quad (57'')$$

Equations (59) and (57″) will give V_2 and T_2. They are to be solved by trial. p_2 and r_2 are fixed by T_2. The work done in the compressor is given by

$$W_c = Mc_v(T_2 - T_1) + m(q_2 + x_2 r_2 - q_1 - x_1 r_1),$$
$$= Mc_v(T_2 - T_1) + m(i_2 - i_1). \quad \ldots\ldots\ldots \quad (62)$$

The remainder of the problem is worked out in the same way as in the previous case.

To apply these formulæ, it is desired to cool a room to 0° with cooling water at 60° F., and the data for 1 ton of refrigeration is to be found. With a 10° rise in the water, a 10° difference between air and cooler and a counter-current air-cooler, the temperature of the air will be reduced to 70° F. The air in the refrigerator will be −10° F. Hence

$$T_1 = -10 + 459.6 = 449.6,$$
$$T_6 = \quad 70 + 459.6 = 529.6.$$

Suppose this to be an open system and p_1 is 14.7 lbs. per square inch absolute, and p_2 is 44.1 lbs. per square inch gauge.

$$T_2 = \left(\frac{58.8}{14.7}\right)^{\frac{1.35-1}{1.35}} 449.6 = 449.6 \times 1.432 = 645° = 185° \text{ F.},$$

$$T_7 = \left(\frac{14.7}{58.8}\right)^{\frac{0.4}{1.4}} 529.6 = \frac{529.6}{1.485} = 356° = -104° \text{ F.}$$

The refrigeration per pound is given by

$$\text{Ref.} = 0.24[450 - 356] = 22.6 \text{ B.t.u.}$$

The weight of air per minute per ton $= \frac{199.2}{22.6} = 8.83$ lbs.

Cooling per minute per ton $= 8.83 \times .24(645 - 530) = 244.$ B.t.u.

Amount of water per minute per ton $= \frac{244}{10} = 24.4$ lbs.

The work with 15% friction is given by

Net work per minute per ton

$$= 8.83 \times 0.24[1.10 \times 1.15(645 - 450) - 0.85 \times (530 - 356)]$$
$$= 209.5 \text{ B.t.u.}$$

Horse-power per ton of refrigeration $= \frac{209.5}{42.42} = 4.95.$

Clearance factor in compressor with 2% clearance

$$= 1 + 0.02 - 0.02 \times (4)^{\frac{1}{1.35}} = 0.964.$$

Clearance factor in expander with 2% clearance

$$= 1 + 0.02 - 0.02 \times (4)^{\frac{1}{1.4}} = 0.966.$$

Displacement per minute per ton for compressor

$$= \frac{8.83 \times 53.34 \times 450}{14.7 \times 144 \times 0.964} = 103.9 \text{ cu.ft.}$$

Displacement per minute per ton for expander

$$= \frac{8.83 \times 53.34 \times 356}{14.7 \times 144 \times 0.966} = 82.2 \text{ cu.ft.}$$

Refrigerating effect

$$= \frac{450 - 356}{1.10 \times 115 \times (645 - 450) - 0.85(530 - 356)} = \frac{94}{99} = 0.95.$$

Dense Air Machine

In the **dense air machine,** or closed system, the initial gauge pressure is taken as 44.1 lbs. per square inch and the pressure p_2 is 220.5 lbs. per square inch gauge. The ratio $\frac{p_2}{p_1}$ is then $\frac{235.2}{58.8}$ or 4. Hence, if computations are made as above there will be no change in any results until the displacement is reached, when it will be found that these quantities are reduced to $\frac{1}{4}$ of their previous values, giving

Displacement per min. per ton for compressor = 26.0 cu.ft.
Displacement per min. per ton for expander = 20.5 cu.ft.

The low performance of the air machine coupled with the large size of the cylinders of open systems has caused this machine to give way to the vapor machines. Since air, however, costs nothing and will not spoil substances nor poison persons if the system leaks, and since machines built for air are reliable, these machines are found in use to a certain extent, especially on vessels.

The **thermodynamics of vapor machines** is best studied from a **temperature-entropy diagram.** This diagram is formed by plotting the **entropy of the liquid** s' against absolute temperature, getting the curve AB which cuts the zero entropy at 32° F., since that is the point from which entropy is measured. The value of s' has been found by plotting the specific heat of liquid c against $\log_e T$ and finding the area to any point.

$$s' = \int_{492}^{T} \frac{cdt}{T} = \int_{492}^{T} cd\,(\log_e T). \quad . \quad . \quad . \quad . \quad (63)$$

For temperatures below 32° F. the value of s' is a negative quantity as is the heat of the liquid q' which is given by

$$q' = \int_{32}^{t} cdt. \quad . \quad . \quad . \quad . \quad . \quad (64)$$

From this curve AB, known as the **liquid line,** the **entropy of vaporization,** $\frac{r}{T}$ is laid off, giving the line CD, the **saturation line.**

If the distance EF is now divided into proportional parts, and this is done with lines at other temperatures and corresponding points are united, the light dotted lines shown in the figure are obtained. These are lines of **constant quality.** On these the quality or the amount of vapor in one pound of vapor and liquid is constant. For that reason these are also called **lines of constant vapor weight.** It is known that the entropy of vaporization for a quality x is

$$\frac{xr}{T}.$$

For this reason the ratio $\frac{EG}{EF}=x$.

From the quality x the volume of 1 lb. of mixture, v, may be found since

$$v=(1-x)v'+xv''. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (65)$$

v = vol. of 1 lb. of mixture;
v'' = vol. of 1 lb. of saturated vapor;
v' = vol. of 1 lb. of liquid.

Now $(1-x)$ is the amount of liquid present in 1 lb., since x is the weight of the vapor. The volume of 1 lb. of liquid is v' cu.ft., and the volume of 1 lb. of dry vapor is v'' cu.ft. In general the quantity $(1-x)v'$ is so small that it may be neglected, giving

$$v=xv''. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (66)$$

The quantities v' and v'' are found in the tables of the properties of vapors, and for any quality x at a given temperature, the volume may be found. Conversely, if the volume be known, the quality at any pressure may be found by

$$x=\frac{v}{v''},$$

and from this equation the values of x for the same volume at different temperatures may be computed, and on joining these points, **lines of constant volume,** shown dotted in the figure, may be drawn.

One other thermodynamic quantity which is of great value

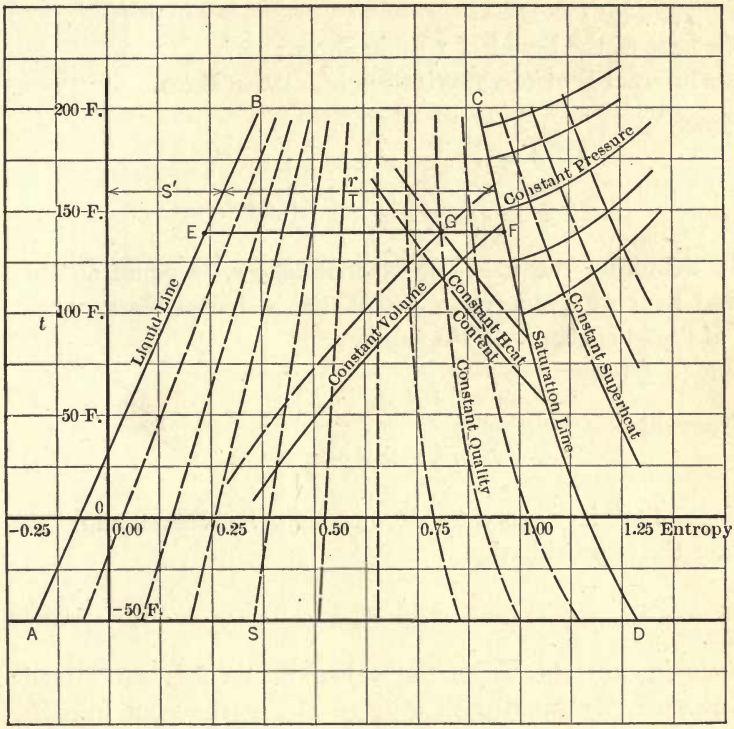


FIG. 24.—*T.S.* Diagram for Ammonia with Thermal Lines.

may be found from the quality x. This is the **heat content** i, which is defined by

$$i = A(u + pv). \quad . \quad . \quad . \quad . \quad . \quad . \quad (67)$$

u = intrinsic energy of 1 lb. of substance;
p = pressure in pounds per square foot;
v = volume of 1 lb. in cubic feet;

$$A = \frac{1}{778}.$$

For a mixture of a liquid and a vapor

$$Au = q' + x\rho. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (68)$$

u = intrinsic energy of 1 lb. of substance in foot-pounds;

$A = \frac{1}{778}$;

q' = heat of the liquid of 1 lb. in B.t.u.;

ρ = internal heat of vaporization of 1 lb. in B.t.u.

Hence

$$i = q' + x\rho + (1 - x)Ap v' + xAp v'',$$
$$= q' + x\rho + xAp(v'' - v') + Ap v'.$$

By definition the **heat of vaporization, r,** is equal to the **internal heat of evaporation** ρ plus the **external work** when 1 lb. of liquid is changed into vapor.

$$r = \rho + Ap(v'' - v').$$

From this

$$i = q + xr + Ap v'. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (69)$$

Since $Ap v'$ is a very small quantity, it is customary to consider the approximation

$$i = q + xr, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (70)$$

as true.

Since in most problems the value of one i is subtracted from another, the small term which is almost the same in each heat content would be canceled out, and for that reason, although an approximation, the use of Eq. (70) would lead to no large errors as it is ordinarily employed.

Since

$$i = q' + xr,$$

$$x = \frac{i - q'}{r},$$

and for any given value of i at any given temperature the quality x may be found and by connecting the points of the

same i at different temperatures, lines of **constant heat content** may be found. These are shown by dotted lines.

The above equations may be written for mixtures:

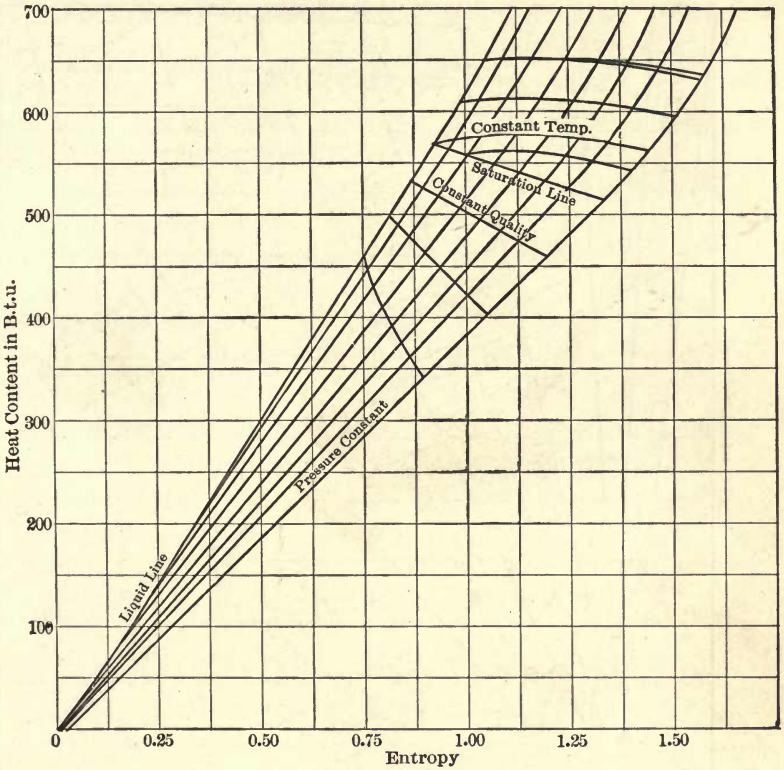


FIG. 25.—Heat Content-Entropy (*I.S.*) Diagram.

$$s = s' + \frac{xr}{T}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (71)$$

$$v = xv'', \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (66)$$

$$i = q' + xr. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (70)$$

For M pounds of substance

$$S = M\left(s' + \frac{xr}{T}\right) \quad \text{or} \quad \Delta S = M(s_2 - s_1), \quad . \quad . \quad (71')$$

$$V = Mxv'' \quad \text{or} \quad \Delta V = M(v_2 - v_1), \quad . \quad . \quad (66')$$

$$I = M(q' + xr) \quad \text{or} \quad \Delta I = M(i_2 - i_1). \quad . \quad . \quad (70')$$

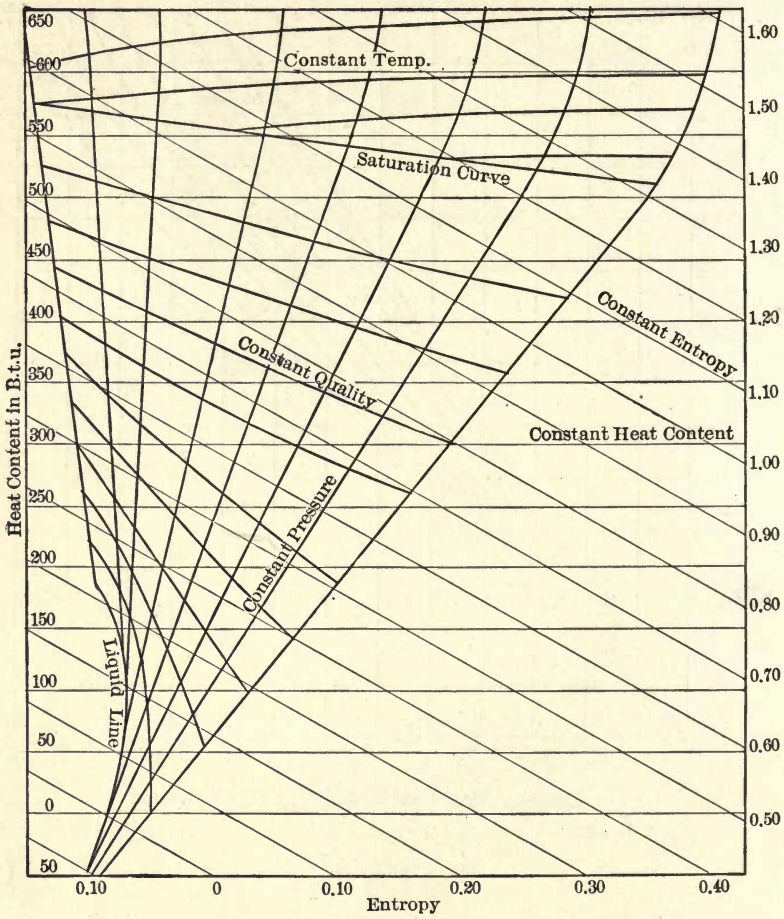


FIG. 26.—*I.S.* Diagram with Inclined Co-ordinates.

It should be remembered that in this saturated region the pressure is fixed for any temperature, by the **characteristic equation** of the vapor, and when the words " any temperature " were used above, the words " any pressure " could have been used.

Since i is an important quantity in most problems relating to refrigeration, and since entropy is needed in discussing adiabatics, the coordinates i and s are often used for a diagram, shown in Fig. 25. This is sometimes called a **Mollier diagram.** In it the line of any temperature or pressure is found by computing the i and s for a given x. If the points of the same quality are united, **lines of constant quality** may be found, while points of the same pressure give **lines of constant pressure.**

This diagram as drawn with the axes of coordinates at right angles is such that for a given range of temperatures, there is much of the figure which is of no value. To correct this the angle between the axes is made much smaller than 90° by some authors and constructors of charts, giving Fig. 26 for the Mollier chart.

If a mixture is heated until all of the liquid is evaporated, the further addition of heat at constant pressure will increase the temperature of the substance above its saturation temperature, or it will superheat it. The difference between the temperature of the substance and its saturation temperature is known as the **degrees of superheat.**

$$\therefore \text{ Deg. superheat} = T_{sup} - T_{sat}.$$

T_{sup} = absolute temperature of the superheated substance;
T_{sat} = absolute temperature of saturation corresponding to the pressure;
$t_{sup} = T_{sup} - 459.6$ = Fahrenheit temperature of the substance;
$t_{sat} = T_{sat} - 459.6$.

If the entropy and heat content are increased, their values will be given by

$$s = s' + \frac{r}{T} + \int_{T_{sat}}^{T_{sup}} \frac{c dt}{T}. \quad . \quad . \quad . \quad . \quad . \quad (71'')$$

$$i = q' + r + \int_{T_{sat}}^{T_{sup}} c_p dt + A pv' = q' + r + \int_{T_{sat}}^{T_{sup}} c_p dt. \quad . \quad (70'')$$

The **specific heat of superheated vapor** is given by the letter c_p. This may be a constant or a variable. In most vapors used for refrigeration, the value of c_p is considered constant for lack of better information. The value of v is found from the **characteristic equation of the superheated vapor,** which is usually taken in the form

$$p(v-c)=BT+ap^{\alpha}. \quad . \quad . \quad . \quad . \quad . \quad (72)$$

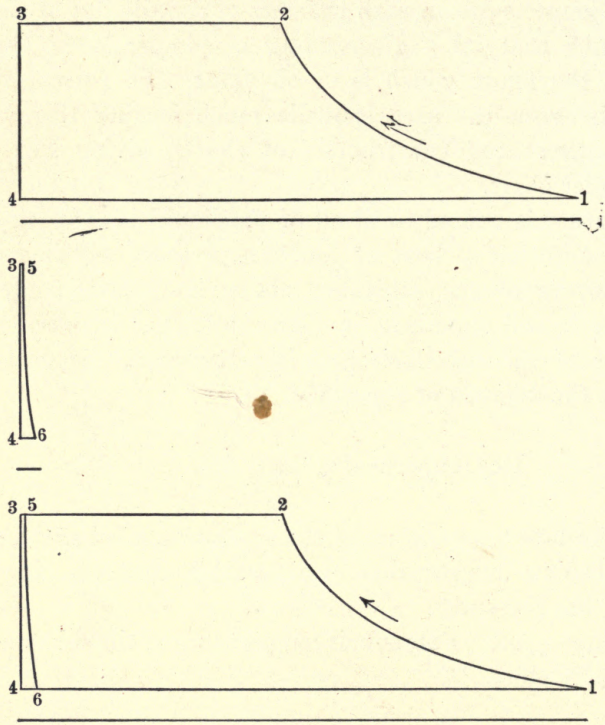


FIG. 27.—Cards from Vapor Machines.

Using these equations, Figs. 24, 25, and 26 are carried out beyond the saturation line into the **superheated region** as shown.

The **p-v diagram** from the vapor machines with an expander is shown in Fig. 27, considering no clearance, as compression and expansion are complete. The vapor is drawn into the

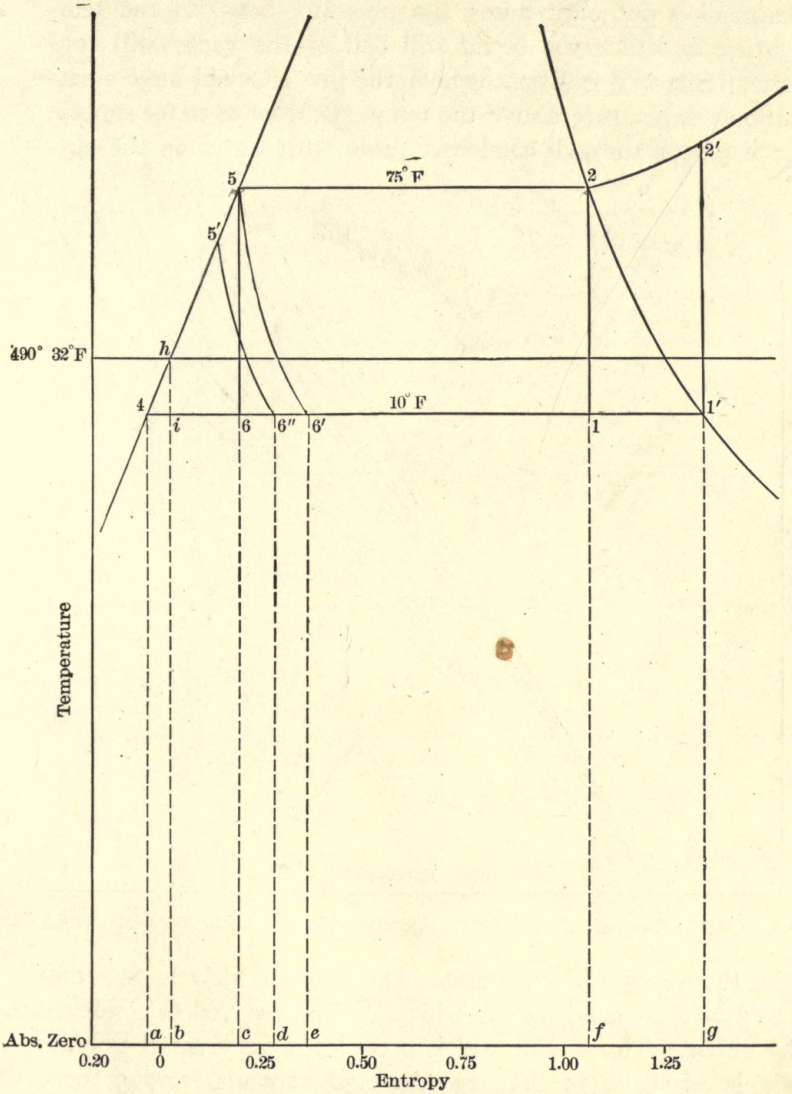


FIG. 28.—*T.S.* Diagram of Cycle of Refrigerating Machine Using a Volatile Fluid.

compressor from 4 to 1 and is compressed from 1 to 2. This compression not only raises the pressure, but also the temperature at which the liquid will boil or the vapor will condense. Hence, if sufficiently high, the pressure will have a saturation temperature above the temperature of a water supply, and if passed through condenser tubes with water on the out-

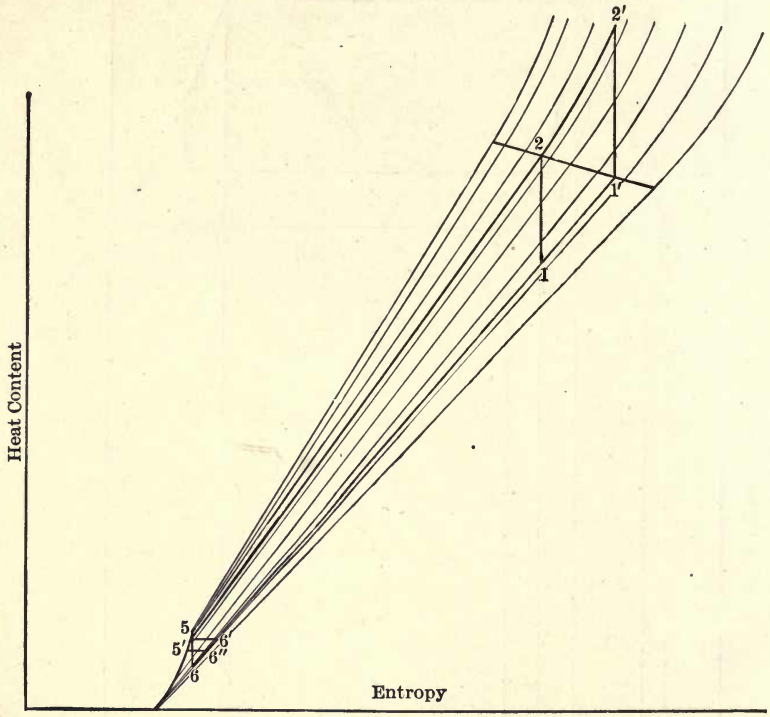


FIG. 29.—*I.S.* Diagram of Cycle of Refrigerating Machine Using a Volatile Fluid.

side, the vapor will condense. The vapor is driven out from 2 to 3 into the condenser in which heat is removed to condense the vapor. The liquid, which occupies a very small volume 3–5, is admitted to the **expander** and expands from 5 to 6, after which it is allowed to enter the expansion coils, where it abstracts heat from the cool material outside of the coils because its pressure has been so reduced that the temperature of boiling is lower than the low temperature of the substance around

the coils. The liquid boils and finally occupies the volume 4–1. The combination of these two cards gives the net card

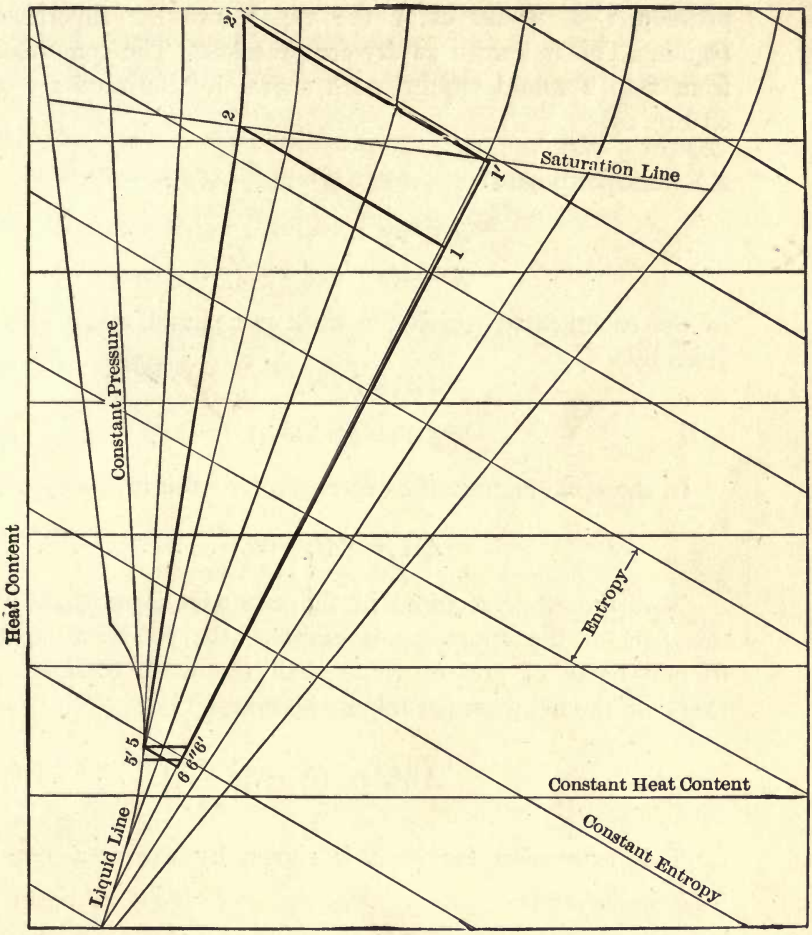


FIG. 30.—*I.S.* Diagram of Cycle of Refrigerating Machine (Inclined Axes).

1–2–5–6, in which 1–2 and 5–6 are adiabatic, and 2–5 and 6–1 are constant-pressure lines.

This figure may be placed on the temperature-entropy diagram, Fig. 28, or on the **i-s** diagram, Figs. 29 and 30.

On the line 1–2 the compression carries the vapor into a

drier region, as drawn $x_2 = 1$, so that the vapor is always saturated and some liquid is present. This is known as **wet compression.** If, however, x_1 as at $1'$ were unity, the compression $1'$–$2'$ would carry the vapor into the superheated region. This is known as **dry compression.** The compression from 1 to 2 would require work shown by the area 1–2–3–4 on Fig. 27.

$$A \times \text{work per pound} = q_2 + x_2\rho_2 - q_1 - x_1\rho_1 + Ap_2v_2 - Ap_1v_1$$
$$= q_2 + x_2\rho_2 + Ax_2p_2v''_2 - [q_1 + x_1\rho_1 + Ax_1p_1v''_1]$$
$$= q_2 + x_2r_2 - (q_1 + x_1r_1) = i_2 - i_1. \quad . \quad . \quad (73)$$

In the superheated region the work per pound would still be given by

$$AW_r = i_2 - i_1,$$
$$AW_r = M(i_2 - i_1). \quad . \quad . \quad . \quad . \quad . \quad (73')$$

In the same manner the expression for work in the expander is

$$AW_e = M(i_5 - i_6). \quad . \quad . \quad . \quad . \quad . \quad . \quad (74)$$

Now the work required in the expander is so slight that this part of the apparatus is omitted, the complication and friction being of greater value than the work regained. In that case the net work per minute becomes

$$AW_n = M(i_2 - i_1)\left(\frac{1}{1-f}\right). \quad . \quad . \quad . \quad (73'')$$

This expression for work is shown by the area 1–2–5–4 on Fig. 28, since,

$$\text{Area } (b\text{–}h\text{–}5\text{–}2) = q'_2 + x_2r_2 = i_2,$$
$$\text{Area } (a\text{–}4\text{–}1\text{–}f) - \text{area } (a\text{–}4\text{–}h\text{–}b) = x_1r_1 + q'_1 = i_1.$$

(q'_1 is a negative quantity.)

$$\therefore \quad \text{Area } (b\text{–}h\text{–}5\text{–}2) - \text{area } (a\text{–}4\text{–}1\text{–}f) + \text{area } (a\text{–}4\text{–}h\text{–}b) = i_2 - i_1$$
$$= \text{Area } (1\text{–}2\text{–}5\text{–}4).$$

On Figs. 29 and 30 there is no area equal to this work, but the difference between the coordinate values of i at 2 and at 1 will give (i_2-i_1) or the work.

In these figures the lines 1–2 represent wet compression, while 1′–2′ represent dry compression.

The pressure is actually reduced from p_2 to p_1 through a throttle valve. Throttling action is of constant heat content. Hence the line 5–6 must be changed to 5–6′, or from a reversible adiabatic or constant entropy line to a non-reversible adiabatic or constant heat-content line. This makes

$$i_{6'}=i_5.$$

The heat removed from the refrigerator is that to pass from 6′ to 1, and hence this heat is

$$\text{Ref.}=M(i_1-i_{6'})=199.2\times\text{tons} \quad . \quad . \quad . \quad . \quad (75)$$
$$=M(i_1-i_5)\ =M\times\text{area}\ (e\text{–}6'\text{–}1\text{–}f).$$

By this equation the quantity M per minute is found.

The heat removed from the ammonia in the condenser is given by the heat under the line 2–5 or 2′–5. This is the area $(c\text{–}5\text{–}2\text{–}f)$. In any case this is given by

$$\text{Cooling}=M(i_2-i_5)=G(q'_o-q'_i), \quad . \quad . \quad . \quad (76)$$

G = lbs. of cooling water per minute;
q'_o = heat of the liquid of cooling water at outlet;
q'_i = heat of the liquid of cooling water at inlet.

These formulæ hold for wet or dry compression. The points 1 and 2 are on the same adiabatic, and hence the values of i_1 and i_2 are found in the same entropy column.

At times after the ammonia is condensed the liquid is passed through pipes surrounded by cold water and the liquid is cooled down the liquid line to 5′. This is known as **after cooling.** This increases the amount of cooling, and if the water is available this gain will increase the refrigeration, as $i_{6'}$ is equal

to i_5 or $i_{5'}$, whichever represents the condition of the liquid entering the throttle valve. It will be seen that throttling action has changed i_6 to $i_{6'}$, thus losing the refrigeration

$$M(i_{6'}-i_6).$$

The quantities may all be found on the I–S diagrams as coordinates of the points. The lines are of peculiar shape. Adiabatics are lines of constant entropy and throttling lines are lines of constant heat content. Constant pressure lines are straight lines in the saturated region, but curve when the superheated region is reached.

The loss of refrigeration due to the elimination of the expander referred to above, and that of the work of the expander are considered to be offset by the simplicity of the apparatus.

The refrigerating effect is given by

$$\text{Ref. effect} = \frac{M(i_1-i_{5'})}{\left(\frac{1}{1-f}\right)M(i_2-i_1)}. \quad . \quad . \quad . \quad . \quad (77)$$

The displacement per minute of the compressor is given by

$$D = \frac{Mxv''}{1+l-l\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (78)$$

$n = 1.2$ for wet compression or 1.33 for dry NH_3.

The clearance factor given in the denominator of this expression is the same as that in any compressor for the lines 1–2 and $1'$–$2'$ are similar lines of the form $pv^n = \text{const.}$ There is no reason why the quality at 2 and that at $2'$ should be different, and if they are the same the lines are the same.

The **volumetric efficiency** of a compressor is the ratio of the free substance actually pumped to the displacement.

$$\text{Vol. Eff.} = \frac{\text{Vol. } V_1}{D} = \left[1+l-l\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\right]\frac{100-\%\text{ leakage}}{100}.$$

Voorhees has patented a scheme by which certain savings may be effected when there is a chance to use two different

temperatures of cooling. In most cases there are certain substances or spaces to be cooled to temperatures not as low as those of other parts of the system, and for such he proposes to expand part of the ammonia to one pressure, and another

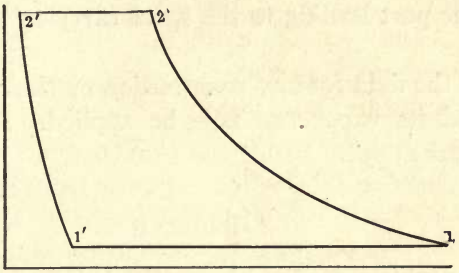


FIG. 31.—Effect of Clearance.

part to a lower pressure. If, now, the compressor is large enough to displace the proper amount of vapor at the lower pressure and at the end of the suction stroke a valve is opened to a place of higher pressure, the spring suction valve to the

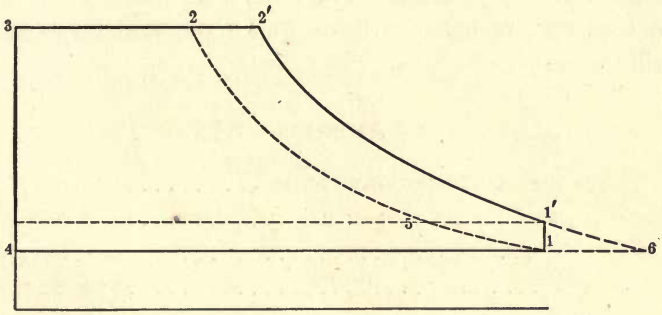


FIG. 32.—Multiple Effect Indicator Cards.

low-pressure place is closed and vapor rushes in from the place of higher pressure and fills the cylinder to that pressure before the return stroke is started. Voorhees calls this a **multiple effect.** The indicator card is shown in Fig. 32. The advantage of this apparatus is that for the part of the heat abstracted

at higher temperature, less work has to be done. 1–2 would be the line for the low-pressure compression. 1′–2′ is the combined line. The work done is 4–1–1′–2′–3. The saving has been 1′–6–1. Voorhees accomplishes this by allowing the piston of the compressor to override a port at the end of the stroke, the port leading to the space carrying the high-pressure vapor.

All of the data for any compression machine using a volatile liquid and its vapor can now be applied. Several problems will be worked out.

Problem

As a problem in the above theory, suppose it is desired to keep a room at 0° F. with water at 60° F., and the amount of refrigeration is to be 1 ton. Investigate for wet and dry compression with ammonia and for wet compression with CO_2 and SO_2.

The temperature of the vapor in the condenser for a 10° rise and a 10° difference will be 80° F. The temperature of boiling with a 5° rise in brine temperature and a 10° difference in temperature between brine and room and brine and vapor will be −25°.

Ammonia—Wet

p_2 corresponding to 80°..... 153.9

x_2..... 1.0

s_2..... 1.0354

i_2..... 557.0

i_3..... 53.6

$i_{3'}$ (if after cooled to 70°)..... 42.1

p_1 corresponding to −25°..... 15.61

$$x_1 = \frac{1.0354 - (-.01290)}{1.3599} \text{.....} \quad 0.858$$

s_1..... 1.0354

$i_1 = q' + xr = -59.8 + 0.858 \times 591.1 = 447.2$

$v_1 = 0.858 \times 16.95$..... 14.5

Refrigeration per lb. of $NH_3 = i_1 - i_{3'} = 447.2 - 42.1 = 405.1$.

Weight of NH_3 per min. per ton of ref. $= \frac{199.2}{405.1} = 0.49$.

Cooling per lb. of $NH_3 = i_2 - i_{3'} = 557.0 - 42.1 = 514.9$.

Lbs. water per lb. $NH_3 = \frac{514.9}{10} = 51.5$ lbs.

H.P. per lb. NH_3 per min. $= \frac{i_1 - i_2}{42.42 \times \text{eff.}} = \frac{557 - 447}{42.42 \times 0.90} = 2.9$.

Volume per lb. of $NH_3 = 14.5$ cu.ft.

Cooling water per min. per ton $= 51.5 \times 0.490 = 25.2$ lbs. per min.

H.P. per ton of capacity $= 2.9 \times 0.490 = 1.42$.

Displacement of piston per min. per ton with no clearance $= 14.5 \times .49 = 7.1$ cu.ft.

Clearance factor with 2% clearance $= 1 + 0.02 - 0.02\left(\frac{153.9}{15.6}\right)^{\frac{1}{1.2}}$

$= 1.02 - 0.135 = 0.885$.

Displacement per min. per ton $= \frac{7.1}{0.885} = 8.0$ cu.ft. per min. per ton.

Refrigerating effect with friction $= \frac{405.1 \times 0.90}{110} = 3.32$.

AMMONIA—DRY

p_1 (−25° F.)	15.6
x_1	1.0
s_1	1.2310
i_1	531.3
v_1	16.95
p_2 (80° F.)	153.9
s_2	1.2310
quality	201° sup.
i_2	680.1
i_3	53.6
$i_{3'}$	42.1

Refrigeration per lb. of $NH_3 = i_1 - i_{3'} = 531.3 - 42.1 = 489.2$.

Weight of NH_3 per min. per ton of ref. $= \frac{199.2}{489.2} = 0.407$.

Cooling per lb. of $NH_3 = i_2 - i_{3'} = 680.1 - 42.1 = 638.0$ B.t.u.

Lbs. water per lb. $NH_3 = \frac{638}{10} = 63.8$ lbs.

H.P. per lb. NH_3 per min. $= \frac{i_2 - i_1}{42.42 \times .90} = \frac{680.1 - 531.3}{42.42 \times .90} = 3.90$.

Volume per lb. of $NH_3 = 16.95$.
Cooling water per min. per ton $= 0.407 \times 63.8 = 25.9$ lbs.
H.P. per ton capacity $= 0.407 \times 3.90 = 1.585$ H.P.
Displacement per min. per ton, no clearance
$= 0.407 \times 16.95 = 6.9$ cu.ft.

Clearance factor with 2% clearance

$$= 1 + 0.02 - 0.02\left(\frac{153.9}{15.6}\right)^{\frac{1}{1.33}} = 0.91.$$

Displacement per minute per ton $= \frac{6.9}{0.91} = 7.59$ cu.ft.

Refrigerating effect $= \frac{489.2 \times .9}{148.8} = 2.96$.

CO_2—Wet

p_1 (at $-25°$ F.)	201.3 lbs.
p_2 (80° F.)	967 lbs.
x_2	1.0
s_2	0.1508
i_2	81.52
s_1	0.1508
$x_1 = \frac{0.1508 + 0.0560}{0.2927}$	0.706
$i_1 = -26.99 + 0.706 \times 127.28$	63.0
$v_1 = 0.451 \times .706$	0.318
$i_{3'} =$	26.21

Refrigeration per lb. of $CO_2 = i_1 - i_{3'} = 63.0 - 26.2 = 36.8$.

Weight of CO_2 per min. per ton of ref. $= \frac{199.2}{36.8} = 5.42$.

Cooling per lb. $CO_2 = i_2 - i_{3'} = 81.52 - 26.21 = 55.31$.

Lbs. of water per lb. of $CO_2 = \frac{55.31}{10} = 5.53$.

H.P. per lb. CO_2 per min. $= \frac{81.52 - 63.0}{42.42 \times 0.80} = \frac{18.52}{42.42 \times .80} = 0.547$.

(80% eff. due to high pressure.)

Cooling water per minute per ton $= 5.42 \times 5.53 = 30.0$.

H.P. per ton capacity $= 0.547 \times 5.42 = 2.96$.

Displacement per min. per ton, no clearance

$$= 5.42 \times 0.318 = 1.72 \text{ cu.ft.}$$

Clearance factor 1% clearance

$$= 1 + 0.01 - 0.01 \times \left(\frac{967}{201}\right)^{\frac{1}{1.3}} = 0.976.$$

Displacement per minute $= \frac{1.72}{0.98} = 1.76$ cu.ft.

Refrigerating effect $= \frac{36.8 \times .8}{18.52} = 2.48$.

This low effect is due to the peculiar properties of CO_2 at the temperature used. At other pressures the properties are such that CO_2 gives a much higher refrigerating effect than other substances. The temperature of 85° F. is not far from the critical temperature of CO_2 and for this reason the results are as above. On the *T–S* diagram of Fig. 33 this is shown clearly by 1–2–5–6′. Plank in the *Zeitschrift für der Gesamte Kälte Industrie* proposes that if the critical temperature is passed, the cooling of the superheated vapor in the condenser should be followed by a further compression, after which a second cooling is resorted to, and that this reduces the entropy so that the state after throttling is changed to give greater refrigeration. The results are shown in the table on the following page.

Condenser Pressure, Atmospheres.	Temperature of CO_2 at Exit from Condenser. Deg. F.	Gain in Refrigeration. Per cent.	Gain in Performance. Per cent.
80	91	48	28
90	101	46	32
100	111	44	35

Fig. 33 shows the compression beyond the critical pressure. The use of the figure is similar, however, to that of the regular

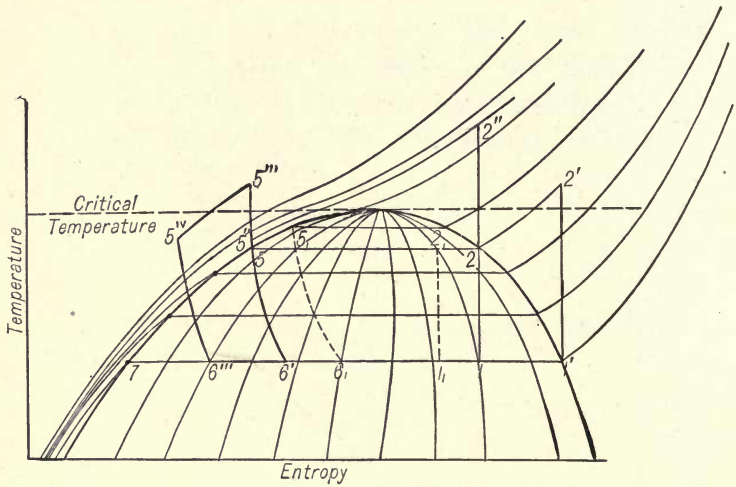


FIG. 33.—CO_2 Diagram on *T.S.* Co-ordinates Showing Plank's Duplex Compression.

T–S diagram. $1–2^{II}–5^{II}–5^{III}–5^{IV}–6^{III}$ is the Plank cycle. The refrigeration under $6'''–6'$ is gained by the compression $5''–5'''$ and the cooling $5'''–5^{IV}$. The cycle $1,–2,–5,–6,$ is one near the critical temperature in which the refrigeration under $6,–1,$ is small when compared with the work $7–1,–2,–5,–7$.

SO_2 WET

p_2 for temperature 80° 59.7
x_2 ..... 1.00
s_2 ..... 0.3020
i_2 ..... 162.08
$i_{3'}$ 12.62

p_1 for $-25°$ 4.98

s_1 ..... 0.3020

$x_1 = \frac{.3020 + 0.0370}{0.4062} =$ 0.835

$i_1 = -17.15 + 176.5 \times 0.835$ 130.1

$v_1 = 14.13 \times 0.835$ 11.8

Refrigeration per lb. of $SO_2 = (i_1 - i_{3'}) = 130.1 - 12.62 = 117.5$.

Weight of SO_2 per min. per ton of ref. $= \frac{199.2}{117.5} = 1.695$.

Cooling per lb. of $SO_2 = i_2 - i_{3'} = 162.08 - 12.62 = 149.46$.

Lbs. of water per lb. $SO_2 = \frac{149.46}{10} = 14.95$.

H.P. per lb. of SO_2 per min. $= \frac{162.08 - 130.1}{42.42 \times .9} = 0.84$.

Cooling water per min. per ton $= 1.69 \times 14.95 = 25.3$.

H.P. per ton of ref. $= 0.84 \times 1.69 = 1.42$.

Clearance factor, 2% clearance

$$= 1 + 0.02 - 0.02\left(\frac{59.7}{4.98}\right)^{\frac{1}{1.26}} = 0.88.$$

Displacement per ton of refrigeration, no clearance

$$= 11.80 \times 1.69 = 20.00.$$

Displacement per ton of refrigeration $= \frac{20.00}{0.88} = 22.7$.

Refrigerating effect $= \frac{117.5 \times 0.9}{31.98} = 3.31$.

With the ammonia system it was seen that 1.42 H.P. is required to produce a ton of refrigeration. If this engine is assumed to use 30 lbs. of steam per H.P. hour and if auxiliaries require 20% of the steam of the main engine, while the boiler evaporates 9 lbs. of steam per pound of coal, the **coal required** would be

$$\frac{1.42 \times 30 \times 24 \times 1.20}{9} = 136 \text{ lbs.}$$

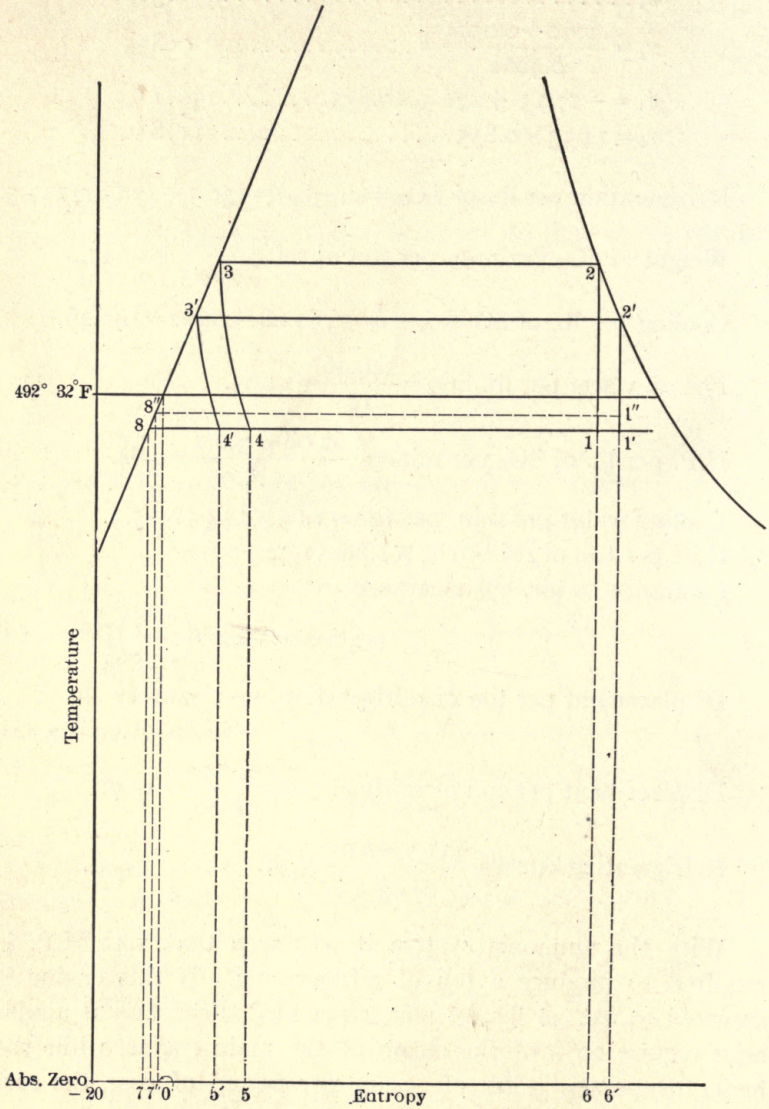


FIG. 34.—Effect of Temperature Range on Refrigerating Effect.

For ice making it takes $1\frac{1}{2}$ to 2 times the tons of ice in refrigeration units, or 272 lbs. of coal would be required per ton of ice. This gives practically $7\frac{1}{2}$ **tons of ice per ton of coal.** This is quite a common figure and one to keep in mind. Of course, as can be seen from the diagrams, the temperature range, or particularly the temperature of the cooling water, plays an important part in the economy of a station. Thus in Fig. 34 if 1234 represent a cycle, the **performance** is $\frac{5\text{–}4\text{–}1\text{–}6}{1\text{–}2\text{–}3\text{–}8}$. If, now, the temperature of the cooling water can be lowered to $2'$–$3'$, the work 1–2–3–8 is decreased to $1'$–$2'$–$3'$–8 and the refrigeration is increased to $5'$–$4'$–$1'$–$6'$. The same effect on work is noticed if the lower temperature can be made higher. Thus, if the back pressure is raised to $1''$–$8''$, then the work is decreased materially and there is an increase in refrigeration. Hence the refrigerating effect is increased. Before telling the refrigerative performance or the cost of producing ice or refrigeration, the temperature range must be known. The smaller the range of temperature the more effective the apparatus. Thus $7\frac{1}{2}$ tons of ice per ton of coal is often reduced to 6 and may reach 15, depending on the temperature ranges. Moreover, if the engine driving the compressor be very efficient, and use 12 lbs. of steam per H.P. hour instead of 30 lbs., the output would be increased very materially. Reports have been made that in Europe results as high as 28 tons of ice per ton of coal have been obtained. This is a matter of temperature range and efficiency of engine. In many cases the steam of the engine is used to give **distilled water** for ice making, and unless evaporators, which complicate the apparatus, are used, a low-grade engine is required to give the necessary amount of water. The usual amount in such plants is 6 to 9 tons of ice per ton of coal. If **raw water** is used, then an engine of greater efficiency may be used, and the result will be increased to 10 or 15. With a gas-producer engine this figure is raised to 20 tons of ice per ton of coal.

The common form of **absorption apparatus** is that using **aqua ammonia.** Before considering the operation of the

apparatus, certain physical phenomena must be noted and quantitative relations given. A solution of 30% NH_3 and 70% water is said to be of 30% **concentration.** The **temperature at which a solution of liquor** of given concentration **will boil,** depends upon the pressure on the liquor. The relation between the pressure and temperature has been determined experimentally by Mollier and plotted in curves and tabulated. The results may be closely approximated in the form of an equation similar to the method used by Macintire.

$$\frac{T_{sat}}{T_{sol}} = 0.00471x + 0.655. \quad . \quad . \quad . \quad . \quad . \quad (80)$$

T_{sat} = temp. of saturated ammonia corresponding to the pressure;

T_{sol} = temp. of boiling;

x = per cent of NH_3 in solution = per cent concentration.

This equation has been derived from the results of experiments of Mollier and Perman as given by Lucke. The tabular values given do not give uniform changes in certain increments, so that there must be some error in these values. The equation will give results within 1% of the tabular values.

When the liquor is heated to drive off the ammonia, both ammonia vapor and water vapor leave the liquor. The **partial water vapor pressure** has been assumed by Spangler to be the steam pressure at the temperature considered, multiplied by the ratio of the number of molecules of water in a certain amount of liquor to the total number of molecules of liquor. Thus

$$\frac{x}{17} = \text{relative number of } NH_3 \text{ molecules,}$$

$$\frac{100-x}{18} = \text{relative number of } H_2O \text{ molecules.}$$

Hence

$$\text{Partial steam pressure} = p\frac{\frac{100-x}{18}}{\frac{x}{17}+\frac{100-x}{18}} = p\frac{1700-17x}{1700+x}, \quad (81)$$

p = steam pressure at given temperature.

The values given by this method of computation agree well with the results of Perman as stated by Lucke.

If 1 lb. of ammonia vapor be absorbed by 200 lbs. of water, it will be found that 893 B.t.u. of heat are developed. This would be the same if more water were used, and it is called the **heat of complete absorption.** If, now, a smaller quantity of water were used, it would be found that less heat would be liberated. This is known as the heat of **partial absorption.** If this solution be added to enough water to make the dilution one in two hundred, the remaining heat of the 893 heat units would be liberated. This is called the **heat of complete dilution.**

Berthelot has found that the heat of complete dilution in B.t.u. is 142.5 times the weight of ammonia per pound of water. This has been checked by Thomsen with a wider range of experiments. This gives, then

$$\text{Heat of complete dilution} = 142.5\frac{x}{100-x}.$$

This gives for the heat of partial absorption

$$q_p = 893 - 142.5\frac{x}{100-x}. \quad . \quad . \quad . \quad . \quad . \quad (82)$$

This is the heat when 1 lb. of ammonia is dissolved in sufficient water to bring the concentration to $x\%$.

If $\frac{x}{100}$ lbs. of ammonia are absorbed,

$$q'_p = 8.93x - \frac{1.42x^2}{100-x}, \quad . \quad . \quad . \quad . \quad . \quad (83)$$

is the heat generated in producing 1 lb. of solution of strength x by adding ammonia to water.

The amount of **water in 1 lb. of mixture** of strength x is $\frac{100-x}{100}$, and the amount of ammonia is $\frac{x}{100}$.

The amount of water is $\frac{100-x'}{100}$, and the ammonia weighs $\frac{x'}{100}$ if the 1 lb. of mixture is of concentration x'.

The amount of ammonia to bring the water in the first case to concentration x' is

$$\frac{100-x}{100}\times\frac{\frac{x'}{100}}{\frac{100-x'}{100}}=\frac{100-x}{100-x'}\frac{x'}{100},$$

or the additional ammonia to change strength from x to x' is

$$\frac{100-x}{100-x'}\times\frac{x'}{100}-\frac{x}{100}=\text{addition of } NH_3. \quad . \quad . \quad (84)$$

The heat produced by this is the difference in the two heats of partial absorption.

$$\text{Heat}=\frac{100-x}{100-x'}\times\frac{x'}{100}\left[893-\frac{142.5x'}{100-x'}\right]-\frac{x}{100}\left[893-\frac{142.5x}{100-x}\right].$$

If this is divided by the weight added, **the heat per pound of NH_3** added to **change the strength from x to x'** is given.

$$q_{pxx'}=893-142.5\left[\frac{x'}{100-x'}+\frac{x}{100-x}\right]. \quad . \quad . \quad (85)$$

When 1 lb. of ammonia is liberated to change the concentration from x' to x, more heat is required than that given above, because the water vapor is driven off with the ammonia and, moreover, the vapors are superheated when driven off from a liquor.

The 893 B.t.u. is the heat developed from the absorption of the vapor. If a liquid is used, 530.7 B.t.u. above 32° F. will have to be added to the heat to care for the generation of the vapor. This would mean that the heat developed

would be less, by this quantity, if liquid ammonia is added to water.

The **specific heat of aqua ammonia** will be taken as unity.

The **specific gravity of aqua ammonia** is given by

$$\text{Sp.gr.} = 1 - \frac{4.3}{1000}\left[x - \frac{x^2}{100} + \frac{x^3}{10{,}000}\right] = S_{60} \quad . \quad (86)$$

for 60° F. For other temperature,

$$S_{60} = S_t + 0.003(t-60)(1-S_t).$$

The **weight of weak liquor to absorb 1 lb. of ammonia vapor** and change its strength from x to x' is given as follows:

$$1 \text{ lb.} = (y+1)\frac{x'}{100} - y\frac{x}{100},$$

$$y = \frac{100-x'}{x'-x}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (87)$$

Problem for Absorption Apparatus

Suppose the water available for cooling is at 60° F., and it is desired to operate the absorption machine with steam at 5 lbs. back pressure.

The temperature of condensation with a 15° rise in temperature of the cooling water and a 10° temperature difference would be 85° F., and this would give an absolute pressure of 167.4 lbs. per square inch for the ammonia. The pressure in the generator to allow for the pressure drop in the separator, rectifier and analyzer would be 169 lbs.

The temperature corresponding to an absolute steam pressure of 20 lbs. is 228° F. Allowing 6° difference in the coils of the generator would give a boiling temperature of 222° F.

The minimum concentration under 169 lbs. and 222° F. would be given by (80):

$$\frac{85.6+459.6}{222+459.6} = 0.00471x + 0.655,$$

$$x = 30.7\%.$$

If higher pressure steam were available, say 25 lbs. gauge, the minimum value of x would be 20.4%.

$$\frac{85.6+459.6}{267.2+459.6}=0.00471x+0.655,$$

$$x=20.0\%.$$

This would increase the ammonia yield and cut down the amount of liquor to be handled.

If the **cold room** is desired at 0° F. with a 10° fall in the brine temperature and 10° temperature differences, the brine would have to be at −20° and the ammonia in the expansion coil would be at −30° F. This means an absolute pressure of 13.56 lbs. per square inch. The absorber pressure would then be a little lower, about 13 lbs. If it is desired to get a solution of 40% concentration in the absorber, the limiting temperature is found as follows:

$$\frac{-31.5+459.6}{T_{sol}}=0.00471\times 40+0.655,$$

$$\frac{429.2}{T_{sol}}=0.843,$$

$$T_{sol}=508=48^\circ \text{ F.}$$

Since it is not possible to maintain this temperature with cooling water at 60° F., a lower concentration must be carried. Assume that a temperature of 80° F. could be maintained. Then

$$\frac{428.1}{80+459.6}=0.00471x+0.655,$$

$$x=29.1\%.$$

It is, of course, impossible to have this, as the absorber would not operate to give the proper concentration.

Suppose then that a 5° rise is permitted in the brine and

5° drop in heat transfer. This would mean a larger brine pump and larger coils. The temperature of the ammonia would then be −15° F., and the pressure would be 20.46 or 20.00 in the absorber.

This gives for 40% concentration.

$$\frac{-15.9+459.6}{T_{sol}}=0.843,$$

$$T_{sol}=527^\circ=67^\circ \text{ F.}$$

This might be possible. It will be required, however, to use 70°, which gives, as the limiting value of x,

$$x=39.1\%.$$

The **amount of weak liquor** allowed to flow per pound of ammonia absorbed is, by (87)

$$y=\frac{100-39.1}{39.1-30.7}=7.25 \text{ lbs.}$$

The **amount of strong liquor** pumped will be 8.25 lbs. per pound of anhydrous ammonia absorbed or evaporated in the refrigerator or expansion coil. This quantity is a little high, due to the fact that the limits of pressure are close. The usual practice is to have between 7 and 8 lbs. pumped per pound of ammonia. From the specific gravity of aqua ammonia, the volume of the liquor for a given amount of refrigeration could be found and from this the displacement of the pump.

It will be well to find the temperature of boiling for a 39.1% solution under the pressure of the analyzer to see if some of the liquor will boil on entering this part of the system. At the top of the analyzer the pressure may be taken as 168.5 lbs. The temperature of boiling is given by

$$\frac{85.4+459.6}{T_{sol}}=0.00471\times 39.1+0.655,$$

$$T_{sol}=650=190.4^\circ \text{ F.}$$

As soon, then, as the strong liquor is heated to 191° F., vapor will begin to come off.

Another computation, which can be made now, is the **possible rise of temperature** of the strong liquor **in passing through the interchanger;** 8.25 lbs. of strong liquor pass one way, while 7.25 lbs. of weak liquor pass the other way.

If the efficiency of the interchanger is assumed to be 90% and the strong liquor is not passed to the rectifier before entering the interchanger, it will be seen that the interchanger will cool off the weak liquor so that there is no need of a weak liquor cooler. The temperature of the weak liquor entering is 222° F., and that of the strong liquor is 70°. The weak liquor is assumed to be cooled to 76° F.

$$0.90[222.0-76]\times 7.25=(t_o-70)8.25,$$

$$t_0=115.5+70=185.5.$$

$$\text{Radiation, } 10\%=105.8 \text{ B.t.u.}$$

Of course, if 76° is assumed to be too high a temperature at which the weak liquor enters the absorber, with water in the coil at 60°, this may be cooled down to 65° in a weak liquor cooler before entering the absorber. In any case this heat will be removed by the cool water in the absorber cooling coil or in this coil and the weak liquor cooler coil together.

The **various conditions** will now be investigated, starting with the **entrance into the condenser.** At this point it is found by some engineers that the best results are obtained if the ammonia is superheated about 20° to 30°. This means that the temperature of the ammonia is 105° F. at this point and that the total pressure is 167.4 lbs. The strength of a solution boiling under these conditions is given by

$$\frac{85+459.6}{105+459.6}=0.00471x+0.655,$$

$$x=65.5\%.$$

By (81) the **partial steam pressure** is

$$p = 1.101 \times \frac{1700 - 17 \times 65.5}{1700 + 65.5} = 0.37 \text{ lb.}$$

The **ammonia vapor pressure** = 167.03 lbs.

The **saturation temperatures** are 71° and 84.9°, given 34° F. of superheat for the water vapor and 20.1° for the NH_3.

The **volume of 1 lb. of superheated NH_3** is 1.89 cu.ft. and the weight of the superheated steam for this volume is about 0.002 lb., a negligible quantity.

For convenience in parts of the problems it will be well to tabulate these conditions in the following form:

CONDITIONS AT ENTRANCE TO CONDENSER

Concentration, x	65.5%
Temperature	105° F.
Pressure steam	0.37
Pressure ammonia	167.03
Saturation temperature, steam	71° F.
Saturation temperature, ammonia	84.9° F.
Superheat, steam	34° F.
Superheat, ammonia	20.1° F.
Specific volume, steam	893 cu.ft.
Specific volume, ammonia	1.89 cu.ft.
Heat content, steam	1108.0 B.t.u.
Heat content, ammonia	572.8 B.t.u.

Weight of water vapor per lb. of ammonia $\frac{1.89}{842} = 0.002$ lb.

If a temperature of 105° F. is used in the rectifier the water coming from the condenser at 70° would be amply cool for this work. The amount of supply must be regulated to bring the temperature to 105°.

The **vapors entering the rectifier** will be assumed to be a little lower than 185.5°, as the rectifier liquor may reduce

this temperature. Take 180° as the first assumption. The pressure here may be 168.5 lbs. and x is given by

$$\frac{85.4+459.6}{180+459.6}=0.00471x+0.655.$$

$$x=41.8\%.$$

The partial steam pressure is

$$p=7.51\frac{1700-17\times41.8}{1741.8}=4.27.$$

CONDITIONS AT ENTRANCE TO RECTIFIER

Concentration	41.8
Temperature	180° F.
Pressure, steam	4.27
Pressure, ammonia	164.23
Saturation temperature, steam	155.5° F.
Saturation temperature, ammonia	83.9° F.
Superheat, steam	24.5° F.
Superheat, ammonia	96.1° F.
Specific volume, steam	89.7 cu.ft.
Specific volume, ammonia	2.29 cu.ft.
Heat content, steam	1140.4 B.t.u.
Heat content, ammonia	621.4 B.t.u.

Weight of water vapor per lb. of ammonia $\frac{2.29}{89.7}=0.026$ lb.

It is seen that there has been some condensation in the rectifier, as the moisture per pound of ammonia at entrance is 0.026 lb., and at exit it is 0.002 lb. This condensation leads to the formation of liquor, as the water will absorb ammonia to 65% concentration. To find the amount of ammonia M entering the rectifier the following equation is used:

$$[(0.026)M-0.002\times1]\frac{.65}{.35}=M-1,$$

$$M=\frac{.9963}{.9516}=1.045 \text{ lb.}$$

The absorption of 0.045 lb. of ammonia will yield $0.045 \times \frac{1.00}{.65} = 0.07$ lb. of solution of 65% concentration and temperature 105°.

The moisture with 1.045 lbs. of ammonia is $1.045 \times 0.026 = 0.027$ lb.

The rectifier has the following **weight balance:**

Weight Balance—Rectifier

Entering:		
Ammonia vapor	1.045	**lbs.**
Water vapor	0.027	
	1.072	lbs.
Leaving:		
Ammonia vapor	1.000	lbs.
Water vapor	0.002	
Liquor	0.070	
	1.072	lbs.

The best way to study the heat required in any part of the apparatus is to find the sum of all the heats entering and leaving. The heat entering with a vapor would be the intrinsic energy plus the work done in forcing the vapor in. The sum of these two would be the heat content. Where there are several vapors together, the sum of the heat contents will give the heat entering or leaving.

For liquids the heat of the liquid must be considered. In addition the heat of absorption must be considered as a negative amount of heat, because the heat generated by this absorption has been removed before the solution reached this point. Heats are figured above 32° F. Absorption experiments are assumed to have taken place at 68° F. Hence the heat content of ammonia at 68° must be added to negative heat of absorption for each pound of ammonia vapor in liquor. $514.7M$ will be called heat of absorbed vapor.

The sum of these will give the total heat at any point.

Thus, if M_a, M_s, and M_l are the weights of ammonia vapor, steam, and liquor, and the strength of the liquor is x, the following is the scheme for the heat at this point:

Energy in ammonia $= M_a i$;

Energy in steam $= M_s i$;

Heat of absorption $= M_l\left(8.93x - \frac{1.42x^2}{100-x}\right)$;

Heat of absorbed vapor $= 514.7 M_e \frac{x}{100}$;

Heat in liquor $= M_l q'$.

The following **heat balance** will then be found:

Heat Balance—Rectifier

Entering:		
Energy in ammonia	$1.045 \times 621.4 =$	649.0
Energy in steam	$0.027 \times 1140.4 =$	30.8
		679.8
Leaving:		
Energy in ammonia	$1.000 \times 572.8 =$	572.8
Energy in steam	$0.002 \times 1108.0 =$	2.2
Heat of liquid of liquor	$0.07 \times (105-32) =$	5.1
		580.1
Heat of partial absorption	$= 0.045\left[893 - \frac{142.5 \times 65}{100-65}\right] =$	28.3
Heat of condensed vapor at atmospheric pressure	$= 0.045 \times 514.7 =$	23.1

Heat removed in rectifier

$$= 679.8 - 580.1 + 28.3 - 23.1 = 104.9 \text{ B.t.u.}$$

Rectifier cooling of water from 68° to 100°

$$= \frac{104.9}{32} = 3.27 \text{ lbs. per lbs. } NH_3.$$

Now, if 8.25 lbs. of liquor of strength 39.1% be mixed with 0.070 lb. of strength 65%, the mixture will be of strength

$$\frac{8.25 \times 39.1 + 0.070 \times 65}{8.32} = 39.3\%.$$

The heat generated by the dilution of the stronger liquor and the concentration of the weaker is given by

$$8.32\left[8.93\times 39.3-\frac{1.42\times\overline{39.3}^2}{100-39.3}\right]-\left\{8.25\left[8.93\times 39.1-\frac{1.42\times\overline{39.1}^2}{100-39.1}\right]+0.070\left[8.93\times 65-\frac{1.42\times\overline{65}^2}{100-65}\right]\right\}$$

$$=2620-(2580+28.6)=11.4 \text{ B.t.u.}$$

The temperature of the mixture is then found as follows:

$$t-32=\frac{8.25\times 153.5+0.070\times 73+11.4}{8.32}=154^\circ \text{ F.}$$

$$t=186.0^\circ \text{ F.}$$

Although 180° was assumed as the temperature at entrance, there will be no recomputation of this point, as the temperature 186.0 may be decreased by radiation. The value 0.9 taken as the efficiency of interchange is not known close enough to warrant recomputing this. The temperature of the liquor of strength 39.3 will be taken as 180° F. This will not begin to boil until it reaches a lower point in the analyzer. Extra interchanger radiation will be $6.0\times 8.32=49.9$. In the analyzer at the upper end there are 1.045 lbs. of superheated vapor leaving with 0.027 lb. of water vapor, and entering at this point are 8.32 lbs. of liquor of 39.3% concentration. At the lower end of the analyzer the conditions of temperature and pressure are as follows:

CONDITIONS AT ENTRANCE TO ANALYZER

Boiling temperature	222° F.
Concentration	30.7%
Pressure, total	169 lbs.
Steam pressure $17.86\frac{1700-17\times 30.7}{1730.7}=$	12.15
Ammonia vapor pressure $169-12.15=$	156.85
Sat. temp. of steam	202.5° F.

Sat. temp. of ammonia	81.1° F.
Steam superheat	19.5° F.
Ammonia superheat	140.9° F.
Volume of 1 lb. of ammonia	2.60 cu.ft.
Volume of 1 lb. of steam	33.0 cu.ft.
Heat content, ammonia	646.2 B.t.u.
Heat content, steam	1157.4 B.t.u.
Weight of steam with 1 lb. of NH_3 $\frac{2.59}{33.0}$ =	0.079 lb.

In passing up through the analyzer the vapor is changed, so that per pound of NH_3 passing there is 0.026 lb. of water vapor present at outlet. The original amount was 0.0785 lb. of vapor per pound of ammonia. This condensation absorbs enough ammonia to make the strength 41.8%. By the method used with the rectifier the amount of ammonia absorbed is 0.04 lb. per pound of ammonia leaving, and the amount of liquor formed is 0.0957. The steam condensed is 0.0557 lb.

The total amount of ammonia at entrance being M, the following holds:

$$M = 1.04M \times 1.045,$$

$$M = 1.087.$$

At entrance there are 1.087 lbs. of NH_3. The liquor formed is 0.100 lb. Hence the liquid dropping back into the analyzer will be

8.25 lbs. of strong liquor of strength 39.1%,
0.07 lb. of strong liquor of strength 65%,
0.100 lb. of strong liquor of strength 41.8%.
This gives 8.420 lbs. of liquor of strength 39.3%.

If there were no evaporation in the analyzer, this liquid would fall into the generator, but because there is heat added to liquor by superheated vapors passing upward some ammonia is driven off. Assume that the temperature of the liquor is raised to 192° F. There must be a balance if this is the case.

CONDITIONS AT 192° F. AT BOTTOM OF ANALYZER

Pressure	169 lbs.
Temperature, assumed	192° F.
Concentration $\left(\frac{85.6+459.6}{192+459.6}=0.00471x+0.655\right)=$	38.8%
Steam pressure $9.75\frac{1700-17\times38.8}{1738.8}=$	5.84
Ammonia pressure $169-5.84=$	163.16
Saturation temperature, ammonia	83.5° F.
Saturation temperature, steam	169° F.
Superheat, ammonia	108.5° F.
Superheat, steam	23° F.
Specific volume, ammonia	2.36 cu.ft.
Specific volume, steam	66.3 cu.ft.
Heat content, ammonia	628.5 B.t.u.
Heat content, steam	1145.5 B.t.u.
Weight of water vapor per lb. of NH_3. . $\frac{2.36}{66.3}=$	0.035

The ammonia set free, M, in changing from 39.3% to 38.8%, is given by

$$(8.42\times0.393-M)\frac{100}{38.8}=8.42-1.035M,$$

$$M=\frac{8.42(0.393-0.388)}{1-(1.035)0.388}=\frac{0.042}{0.598}=0.070.$$

The amount of water vapor leaving is

$$0.070\times0.035=0.002.$$

The amount of liquor falling into the generator equals

$$8.42-0.070-0.002=8.348 \text{ lbs.}$$

The amount of ammonia vapor coming from the generator amounts to

$$1.087-0.070=1.017 \text{ lbs.}$$

The water vapor leaving amounts to

$$1.017 \times 0.079 = 0.080 \text{ lb.}$$

The weak liquor left in the generator is equal to

$$8.348 - 1.097 = 7.251.$$

This should mount to 7.248, since 0.002 lb. of water enters the absorber with the ammonia.

Weight Balance for Analyzer

Entering:

From generator	ammonia	1.017	
	water vapor	0.80	
From rectifier, liquor		.070	
From interchanger, liquor		8.250	
			9.417

Leaving:

To rectifier	ammonia	1.045	
	water vapor	0.027	
To generator, liquor		8.348	
			9.420

Heat Balance for Analyzer

Entering:

From generator	ammonia. $1.017 \times 646.2 =$	656.0
	steam. . . $0.080 \times 1157.4 =$	92.6
From rectifier and interchanger		
Heat of liquid of liquor $8.320 \times (180 - 32) =$		1230.0
		1978.6

Heat of partial absorption,

$$= -8.320\left[8.93 \times 39.3 - \frac{1.42 \times \overline{39.3}^{2}}{60.7}\right]$$

$$= -2620 = -2620$$

Heat of absorbed vapor

$$8.32 \times 0.393 \times 5.147 = 1675$$

Leaving:

To rectifier { ammonia. . 1.045 × 621.4 = 650
steam . . . . 0.027 × 1140.4 = 30.9

To generator,

Liquor, heat of liquid $8.348[192-32] = 1335.0$

2015.9

Heat of partial dilution

$$-8.348\left[8.93\times38.8-\frac{1.42\times\overline{38.8}^{2}}{61.2}\right]=-2600$$

Heat of absorbed vapor

$$8.348\times0.388\times514.7=1662$$

Heat to drive off ammonia $2620-2600=20$ B.t.u.

Excess leaving. . . . . $2015.9-1978.6=+37.3$ B.t.u.

Heat of atm. pressure . . . . $1662-1675=-13$ B.t.u.

In other words there is an excess of 44.3 B.t.u. and consequently the liquor cannot be warmed to 192°.

Try 188°.

Pressure. . . . . 169 lbs.
Temperature assumed. . . . . 188° F.
Concentration. 39.9

This is stronger than the original liquor, so there can be no evaporation.

Try 190°.

Pressure. 169
Temperature assumed. . . . . 190°
Concentration. 39.3

This is possible, as the liquor is just heated to its limit. Hence all will fall into the generator giving as the weight balance the following:

Weight Balance for Analyzer

Entering:

From generator,	
Ammonia	1.087
Water vapor	0.086
From rectifier and interchanger,	
Liquor	8.320
	9.493

Leaving:

To generator, liquor (8.32+0.096)		8.420
To rectifier	ammonia	1.045
	steam	0.027
		9.492

Heat Balance

Entering:

From generator	ammonia. 1.087×646.2 =	703.0
	steam. . 0.c86×1157.4 =	99.6
From rectifier and exchanger, liquor		
	8.32×(180−32) =	1230
		2032.6
Heat of partial absorption	=	−2620.0

Heat of absorbed vapor $8.32 \times 0.393 \times 514.7 = 1685$

Leaving:

To generator, liquor. . . . . 8.42(190−32) =		1330.0
To rectifier	ammonia. . . . 1.045×621.4 =	649.0
	steam. . . . . 0.027×1140.4 =	30.8
		2009.8

Heat of partial absorption $\frac{8.42}{8.32} \times 2620.0 = -2650$

Heat of absorbed vapor $8.42 \times 0.393 \times 514.7 = 1700$

There are 22.8 B.t.u. entering in excess and there are 30 B.t.u. given off to care for the heat in large amounts of liquor. To allow for condensation of vapor above atmospheric pressure, there will be an excess of 15 B.t.u. This gives 37.8 B.t.u. in excess. This would raise the liquor 5 degrees, but this is

impossible as 192° is too high. Suppose 190.8° is tried. Since at 192 there are 42.3 B.t.u. in excess leaving, and at 190 there are 37.8 B.t.u. in excess entering.

Pressure	169
Temperature	190.8
Concentration	39.1
Steam pressure $\ldots\ldots 9.50\frac{1035}{1739} =$	5.65
Ammonia pressure $\ldots\ldots 169 - 5.65 =$	163.35
Saturation temperature, ammonia	83.5° F.
Saturation temperature, steam	167.5° F.
Superheat, ammonia	107.3° F.
Superheat, steam	23.3° F.
Specific volume, ammonia	2.35
Specific volume, steam	78.00
Heat content, ammonia	628.0
Heat content, steam	1144.9
Lbs. of water vapor per lb. of $NH_3 \frac{2.35}{78.0} =$	0.030
NH_3 set free $\ldots\ldots 8.42\left(\frac{(0.393-0.391)}{1-1.030\times 0.391}\right) =$	0.028
Steam set free $\ldots\ldots 0.028\times 0.030 =$	0.001

Weight Balance of Analyzer (3d Assumption)

Entering:	
From generator,	
Ammonia $\ldots\ldots 1.085 - 0.028 =$	1.057
Steam $\ldots\ldots 0.086 - 0.001 =$	0.085
From rectifier and interchanger,	
Liquor	8.320
	9.462
Leaving:	
To generator, liquor $\ldots\ldots 8.420 - 0.029 =$	8.391
To rectifier { ammonia $\ldots\ldots =$	1.045
To rectifier { steam $\ldots\ldots =$	0.027
	9.463

Heat Balance for Analyzer

Entering:

From generator,

Ammonia.....1.057 = 646.2 = 683.0

Steam.....0.085 × 1157.4 = 98.5

From rectifier and interchanger, liquor... = 1230

2011.5

Heat of partial absorption

$$8.32\left[8.93\times39.3-\frac{1.42\times\overline{39.3}^{2}}{60.7}\right]=-2020$$

Heat of absorbed vapor

$$8.32\times0.393\times514.7=1680$$

Leaving:

To rectifier,

Ammonia.....1.045 × 621.4 = 649

Steam.....0.027 × 1140.4 = 30.8

To generator, liquor....8.391(192.8 − 32) = 1330.0

2009.8

Heat of partial absorption

$$-8.391\left[8.93\times39.1-\frac{1.42\times\overline{39.1}^{2}}{60.9}\right]=-2620$$

Heat of absorbed vapor

$$8.391\times0.391\times514.7=1683$$

Heat of absorbed vapor.....1683 − 1680 = 3 B.t.u.

Heat of concentration.....2620 − 2620 = 0 B.t.u.

Excess heat leaving.....2009.8 − 2011.5 = −2.7 B.t.u.

Excess heat, entering..... = 0.3 B.t.u.

If 190 gave 37.8 B.t.u. excess entering and 190.8 gave 0.3 B.t.u. excess entering, the value of 190.85° is probably correct. It is not worth working as close as this and the 37.8 B.t.u. excess entering may be assumed to be cared for by radiation, giving the second computation as the one required.

The investigation of the generator now follows.

Weight Balance for Generator

Entering:

From analyzer, liquor	8.420 lbs.
	8.420 lbs.

Leaving:

To analyzer,	
Ammonia	1.087
Steam	0.086
To exchanger,	
Liquor	7.248
	8.421

Heat Balance for Generator

Entering:

From analyzer,	
Liquor $8.420(190-32)=$	1330
Heat of partial absorption $=-2650$	
Heat of absorbed vapor	
$8.416\times0.393\times514.7=$	1700

Leaving:

To analyzer,	
Ammonia $1.087\times646.2=$	703
Steam $.086\times1157.4=$	99.6
To interchanger,	
Liquor $7.248[222.0-32]=$	1378.0
	2180.6
Heat of partial absorption	
$7.248\left[8.93\times30.7-\frac{1.42\times\overline{30.7}^{2}}{69.3}\right]=$	−1845
Heat of absorbed vapor	
$7.248\times.307\times514.7=$	1149
Heat for difference in heats of partial	
absorption $2650-1845=$	805
Heat excess in leaving $2180.6-1330=$	850.6
Heat in atm. pressure $1149-1700=$	− 551.0
	1104.6

Pounds of exhaust steam at 20 lbs. absolute pressure, of quality 0.85 required to produce this heat is given by

$$\text{Lbs. of steam} = \frac{1104.6}{.85 \times 961.7} = 1.35 \text{ lbs.}$$

If 10% radiation is assumed the steam will be 1.5 lbs.

At the discharge of the condenser at 85° F., the pressure is 167.4 lbs. and the strength of the solution that can be formed is, by (80),

$$1 = 0.00471x + 0.655,$$

$$x = 73\%.$$

This result is large, and, moreover, the equation is not true for more than 50% concentration. The condition of the liquor is not known. The quantity formed in any case is not large, so it will be assumed that the strength is 70%, and hence on the condensation of 0.002 lb. of water, the liquor formed will be 0.005 lb. This gives the following weight balance.

Weight Balance for Condenser

Entering:	
From rectifier,	
Ammonia	1.000
Steam	0.002
	1.002
Leaving:	
To throttle valve,	
Liquid NH_3	0.997
Liquor	0.005
	1.002

The heat balance is as follows:

Heat Balance for Condenser

Entering:

From rectifier,

Ammonia.....1.000×572.8 = 572.8

Steam.....0.002×1108.0 = 2.2

575.0

Leaving

To throttle valve,

Liquid ammonia.....0.997×59.4 = 59.2

Liquor.....0.005(85−32) = 0.3

Heat of partial absorption

$$0.003\left[893-\frac{142.5\times 70}{30}\right]=-1.7$$

Heat of absorbed vapor...0.003×514.7 = 1.5

59.3

Heat removed.....575.0−59.3 = 515.7 B.t.u.

Lbs. of water heated from 60 to 75° per lb. of ammonia entering condenser

$$\frac{515.7}{15}=34.4 \text{ lbs.}$$

(If water from 68° to 75° is used, the amount required will be 73.7 lbs., or the water from the absorber could be used in the condenser.)

EXPANSION OR THROTTLE VALVE AND EXPANSION COIL

NOTE.—No radiation is assumed from receiver.

This action is constant heat content action. Hence i after expansion is that for the liquid at 85° or 59.4 B.t.u. The heat content for dry ammonia at −15° is 534.3 B.t.u. Hence the refrigeration produced is equal to

$$0.997(534.3-59.4)-0.005[85-(-15)]=472.9.$$

If 10 B.t.u.'s are assumed for leakage, this gives 462.9 B.t.u. of heat abstracted in the expander per pound of ammonia entering the condenser.

The number of pounds of ammonia per minute per ton of refrigeration is

$$\frac{199.2}{462.9} = 0.430 \text{ lb.}$$

Absorber

Weight Balance for Absorber

Entering:

From expander,

Ammonia	0.997	
Liquor	0.005	
From interchanger,		
Liquor	7.248	
		8.250

Leaving:

To interchanger,		
Liquor	8.250	
		8.250

Heat Balance of Absorber

Entering:

From expander,

Ammonia $\ldots\ldots 0.997 \times 534.3 = 532.8$

Liquor $\ldots\ldots 0.005 \times (-15-32) = -0.2$

Heat of partial absorption

$$0.003\left[893 - \frac{142.5 \times 70}{30}\right] = -1.7$$

Heat of absorbed vapor $\ldots\ldots = 1.5$

From interchanger,

Liquor $\ldots\ldots 7.248 \times (76-32) = 319.0$

Heat of partial absorption $\ldots\ldots = -1845$

Heat of absorbed vapor $\ldots\ldots$ 1149

Leaving:

To interchanger,

Liquor $\ldots\ldots 8.25 \times (70-32) = 313$

Heat of partial absorption

$$8.25\left[8.93 \times 39.1 - \frac{1.42 \times \overline{39.1}^{2}}{60.9}\right] = -2580$$

Heat of absorbed vapor

$$8.25 \times 0.391 \times 514.7 = 1658$$

Heat entering − heat leaving

$$= [532.8 - 0.2 + 319 - 313]$$
$$+ [2580 - 1845 - 1.7] - [1658 - 1149 - 1.5]$$
$$= 538.6 + 733.3 - 507.5 = 764.4 \text{ B.t.u.}$$

If 60° water enters and is raised to 68° F. in the cooling coil, the cooling water required will be

$$G = \frac{764.4}{8} = 95.42 \text{ lbs. per lb.}$$

This is excessive and in practice a greater range of temperature would be used. This would reduce the quantity.

Work of Pump

The work done in the pump is given by

$$\text{Work} = \frac{pV}{778}$$
$$= \frac{(169 - 20.00)144 \times 8.25}{778 \times 62.4\left[1 - \frac{4.3}{1000}\left(39.1 - \frac{39.1^2}{100} + \frac{39.1^3}{10{,}000}\right)\right]}$$
$$= 4.18 \text{ B.t.u.}$$

This heat is added by the pump and is cared for by radiation. The following heat balance is made:

	Heat Added.	Heat Taken Away.	Radiation.
Generator	1104.6	.....	.....
Analyzer	.....	.....	37.8
Rectifier	.....	103.1	.....
Condenser	.....	515.7	.....
Expander	462.9	.....	−10.0
Absorber	.....	764.4	.....
Pump	4.2	.....	.....
Interchanger	.....	.....	105.8 4.2 (pump) 49.9 (p. 91)
	1571.7	1383.2 187.7 1570.9	187.7

If $1\frac{1}{2}$ tons of refrigeration are required per ton of ice, this apparatus would require

$$1.5 \times 0.43 \times 24 \times 60 = 930 \text{ lbs.}$$

of ammonia entering the condenser per day.

The steam needed for this would be $930 \times 1.4 = 1305$ lbs. This would require the exhaust of a 21 H.P. engine to supply the steam

$$\left(\frac{1305}{24 \times 30} = 21\right).$$

If the steam were supplied by a boiler, the coal required would be $\frac{1305}{10} = 130.5$ lbs. The ice per pound of coal would be

$$\frac{2000}{130.5} = 15.3 \text{ lbs.}$$

In practice these plants yield from 9 to 10 tons of ice per ton of coal supplied. In a test by N. H. Hiller, 60 tons of ice required 3890 lbs. of steam per hour. Assuming that this high pressure steam is made at the rate of 8 lbs. per pound of coal, the coal required for 60 tons of ice would be 11,670 lbs. or 5.8 tons. This gives a value of 10.3 tons of ice per ton of coal. The steam used for this apparatus could have been the exhaust steam from an engine, and consequently the full coal should not be charged to ice-making.

To give some comparative figures from the problems in this chapter, the results have been collected in the following table:

	Entrance temp. to Compressor.	Leaving temp. from Cooler.	Absolute Upper Pressure.	Absolute Lower Pressure.	Displacement per Minute per Ton Compressor.	Displacement per Minute per Ton Expander.	Horse Power to Drive Machine.	Water per Minute Per Ton.	Ref. Eff.
	F.	F.						Lbs.	
Air, atmospheric	−10	70	58.8	14.7	103.9	82.2	4.95	24.4	0.95
Air, dense	−10	70	235.2	58.8	26.0	20.5	4.95	24.4	0.95
Ammonia, wet	−25	80	153.9	15.6	8.0	...	1.42	25.2	3.32
Ammonia, dry	−25	80	153.9	15.6	7.0	...	1.58	25.9	2.96
CO_2, wet	−25	80	967.0	201.3	1.76	...	2.96	30.0	2.48
SO_2, wet	−25	80	59.7	5.0	22.7	...	1.43	25.3	3.31

The **refrigerant** to be used is determined by the designer of the plant. Each has certain advantages. **Air** is the cheapest of all, but its properties are such that large displacements are necessary, even with dense-air machines, and for ordinary temperature ranges the refrigerating effect is small. **Sulphur dioxide** and **ammonia** are objectionable on account of danger to life and property in case of breaks in the system. **Carbon dioxide** is not objectionable from this cause. The CO_2 and SO_2 are much cheaper than ammonia; when the pressure range is considered it is found that carbon dioxide requires excessive pressures on both sides of the system, thus necessitating steel cylinders, special packings, heavy piping and fittings, but a small size compressor. The pressures with sulphur dioxide are not great and with ammonia, although the pressure is high on the upper side, it is not so high as to require special constructions. The sulphur dioxide compressor is large as compared with the ammonia compressor. Carbon dioxide is near the critical temperature at ordinary water temperatures, and this causes certain changes to be made. As was shown on Fig. 33, this substance may be operated above the critical point. Ammonia is the most common substance employed, but there is a tendency to use carbon dioxide to a greater extent than formerly.

Experimental runs have shown that these substances give about the same practical results. The SO_2 and NH_3 corrode metals slightly and CO_2 and SO_2 machines being nearer than NH_3 to their critical temperatures, will not cause excessive pressure if the condenser-water should fail, as has happened with NH_3, causing rupture in the system. NH_3 with oil forms a combustible, which cannot be said of SO_2 and CO_2. Carbon dioxide can be brought in contact with any metal, while NH_3 and SO_2 must be kept in contact with iron and steel only.

Mixtures of CO_2 and SO_2 have been tried.

Methyl and ethyl alcohol and methyl chloride have been used as refrigerants.

In refrigeration, 2 gals. of water per minute are generally required per ton of refrigeration.

The effect of temperature range is seen by the following table, given by Thomas Shipley in the Bulletins of the York Manufacturing Co.:

VOLUMETRIC EFFICIENCY, DISPLACEMENT PER MINUTE PER TON AND COMPRESSOR HORSE-POWER PER TON FOR YORK SINGLE-ACTING COMPRESSOR.

High Pressure by Gauge and Temp. of Saturation.	Suction Pressure and Temperatures of Saturation.								
	5 Lbs. Gauge or −17.5° F.			10 Lbs. Gauge, or −8.05° F.			15.67 Lbs. Gauge or 0° F.		
	Vol. Eff.	Disp. per Min. Cu.ft.	I.H.P.	Vol. Eff.	Disp. per Min. Cu.ft.	I.H.P.	Vol. Eff.	Disp. per Min. Cu.ft.	I.H.P.
145 lbs., 82° F.....	0.79	7.28	1.65	0.812	5.7	1.4	0.83	4.5	1.2
165 lbs., 89° F.....	0.775	7.5	1.83	0.797	5.9	1.56	0.815	4.6	1.34
185 lbs., 95.5° F....	0.76	7.8	2.01	0.782	6.0	1.72	0.80	4.8	1.49
205 lbs., 101.4° F...	0.745	8.05	2.19	0.767	6.2	1.89	0.785	5.0	1.63

High Pressure by Gauge and Temp. of Saturation.	Suction Pressures and Temperatures of Saturation.					
	20 Lbs. Gauge or 5.7° F.			25 Lbs. Gauge or 11.5° F.		
	Vol. Eff.	Disp. per Min. Cu.ft.	H.P.	Vol. Eff.	Disp. per Min. Cu.ft.	H.P.
145 lbs., 82° F.....	0.842	3.9	1.06	0.855	3.4	0.94
165 lbs., 89° F.....	0.827	4.1	1.20	0.84	3.5	1.07
185 lbs., 95.5° F....	0.812	4.2	1.34	0.825	3.6	1.20
205 lbs., 101.4° F...	0.797	4.3	1.47	0.81	3.7	1.32

From the above table it is seen that the range of temperature has a great effect, the variation in the table being from 2.19 H.P. per ton to 0.94 H.P. per ton. It is absolutely necessary to know conditions before a given problem can be solved. The smaller the range of temperature, the less the H.P. required. The table has been based on tests made on compressors with a clearance of not more than $\frac{1}{32}''$ and with no after cooling. With after cooling there would be a reduction in horse-power. To find the engine horse-power, an allowance of 17% must be added for small compressors, and 15% for large compressors to care for friction.

To utilize the fact that small temperature range means

an increase of efficiency, Mr. G. T. Voorhees patented the application of **multiple effect to absorption and compression machines,** when different temperatures are applicable on the lower side of the system. The compressor system and absorber system are shown in Fig. 35. In this system the vapor from the compressor or rectifier is sent to the condenser and after it passes a throttle valve to reduce its pressure to a point above

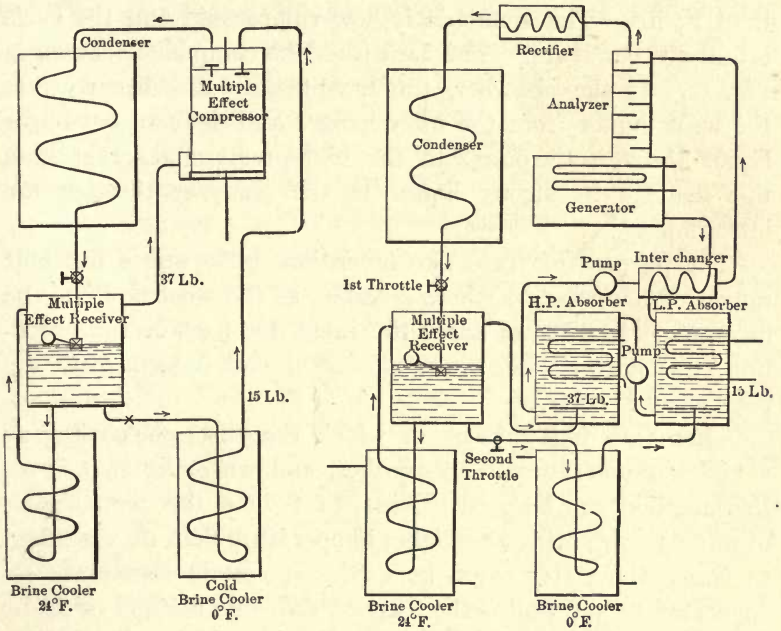


FIG. 35.—Voorhees Multiple Effect Apparatus.

the lowest pressure used, it is caught in a receiver called by Voorhees, **a multiple effect receiver.** From the receiver some of the liquid may pass without throttling to a brine cooler, and the evaporation from this passes back into the receiver. Some of the liquid from the receiver is passed through another throttle valve and is delivered at a lower pressure to another brine cooler or refrigerating coil. The low pressure in this coil is maintained by a compressor or by an absorber of low pressure. In the case shown in the figure, the absolute pres-

sure is about 15 lbs. per square inch. The vapor formed in the receiver from the evaporation in the first cooler and from the throttling of the liquid in the first expansion valve is taken to an absorber of 37 lbs. absolute pressure in one case, or to a cavity at the end of the stroke of the compressor, so that when the piston overrides a port at the end of the stroke, this vapor at 37 lbs. pressure will flow in, since the pressure inside of the compressor at the end of the suction stroke is slightly less than 15 lbs. The ammonia is now compressed and the cycle is followed out again. The card from the compressor is seen in Fig. 32. In the absorber, the low-pressure absorber receives the weak liquor from the interchanger and delivers a stronger liquor through the pump to the high-pressure absorber, and this delivers its strong liquor to the analyzer through the interchanger.

The purpose of these two inventions is to utilize different ranges of temperature where possible, as the efficiency may be increased. In cutting down the range for part of the operation, this part is done more efficiently and consequently the total effect is better.

There are many cases in which there is some cooling at a higher temperature than another, and whenever that is so, this method can be used. Thus, to reduce the temperature of water to 40°, brine at a higher temperature than that required to freeze the water could be used. If certain rooms are refrigerated to 35° while others are at 20°, this method could be employed. One of the latest applications is by the Quincy Market Cold Storage Co., of Boston, in their new 1000-ton compressor, the largest ever built. This compressor draws ammonia from two systems, the cold-storage system of low pressure and the conduit system at a higher pressure.

CHAPTER IV

TYPES OF MACHINES AND APPARATUS

THE compressor is the important part of refrigerating apparatus. Fig. 36 shows a section through the housing of a York compressor. The driving of these compressors is accomplished by an engine or electric motor. The former is shown in Fig. 14. Fig. 36 shows a section through both cylinders. As is true in most large vertical compressors, the two compressors are connected to a common shaft with cranks at right angles. One steam cylinder is usually employed. This is horizontal and is connected to one of the two crank pins. The form of housing is clearly indicated by Figs. 14 and 36. The housings must be solid and of proper section to make a rigid construction. The box-girder columns are connected at the bottom by the casting which carries the main bearing and gives a very strong form. The working platforms for large machines are carried from the housing or frame. The fly-wheel is placed between the cylinders, being supported in a simple manner by the two bearings.

The cylinders are built as shown in the figure. They are single acting and are made of close-grain metal. The suction enters at the bottom of the cylinder on the up-stroke of the piston. The piston is made long and has four piston rings to give tightness. The piston casting is carried by a **spider** and **hub** so that the ammonia may pass through the suction valve at the center of it. This valve has a central spindle to which is attached a **cushion head.** This is a plate. A projecting cup carried by a spider from the removable seat receives the plate. This plate and cup fit so closely that they form a dash pot and prevent the hammering of the valve and limit its motion. A spring is also used to aid in supporting the valve.

The gas is sucked into the upper end on the down-stroke of the piston, and when the compressor reverses, the valve is closed and the ammonia is compressed until the pressure

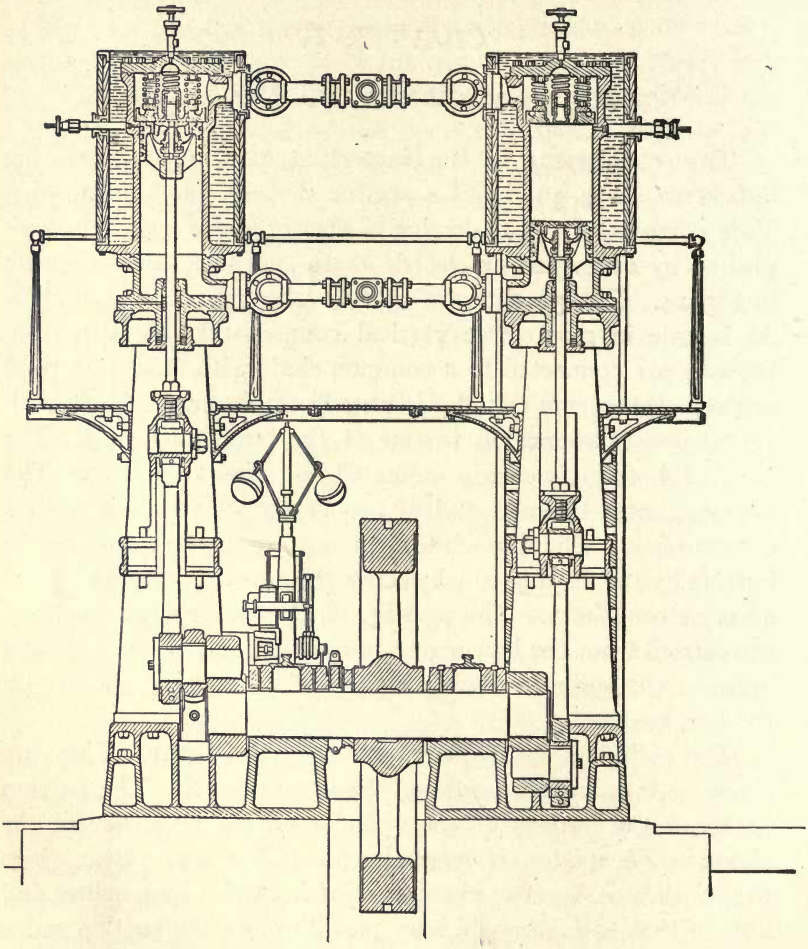


FIG. 36.—Cross-section of York Compressor.

beneath the valve at the center of the head is greater than the pressure above. As will be seen, this valve is controlled by a small spring on top of it, but the dash pot into which a cylindrical projection on the back of the valve fits, pre-

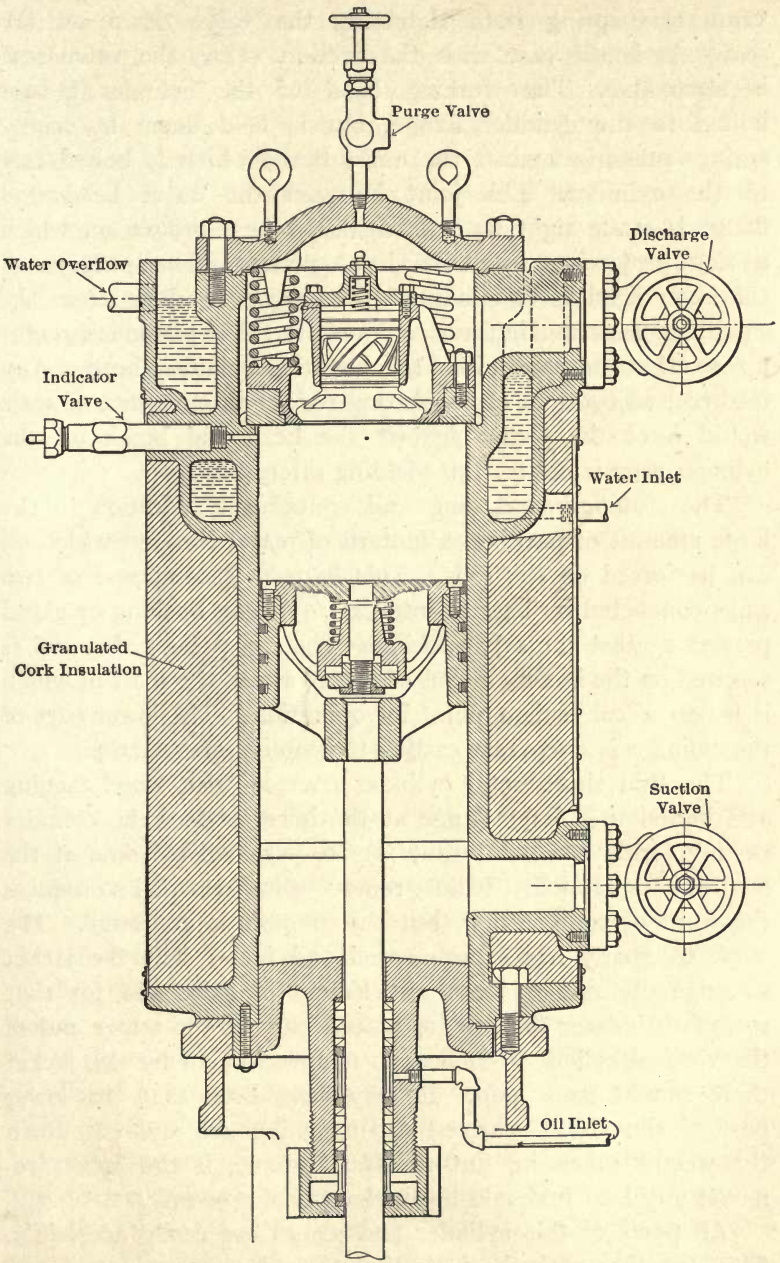


FIG. 36*a*.—Sectional View of York Compressor.

vents this spring from slamming the valve down on the seat. As is the case with the suction valve, the valve seat is removable. The working head of the cylinder is not bolted to the cylinder flange, but is held down by heavy springs pressing against the outer head, which is bolted fast to the cylinder. The joint between the outer head and flange is made tight by a lead gasket in a groove on which a ring projecting from the head presses. The purpose of the inner head is to eliminate the danger resulting from the small clearance used in these compressors. The piston is brought up so that it practically touches the cylinder head. Any incorrect adjustment of connecting rod, or the presence of scale would force the piston against the head and break off the cylinder were it not for this yielding safety head.

The stuffing-box is long and contains, in addition to the large amount of packing, a **lantern** of metal through which oil can be forced on the rods. This lantern is composed of two rings connected by bars at intervals. A long bushing or gland presses against the soft packing. The cap presses this and is screwed on the box by means of a gear wheel, the shaft of which is led to a convenient point for operation. The lower part of the cylinder is a separate casting to simplify construction.

The thin sheet-metal cylinder covered with wood lagging and bolted to a large flange at the lower end of the cylinder casting forms a water jacket for the removal of some of the heat of compression. It does remove some heat and so reduces the work of compression, but the amount is not large. The York Company has experimented and found that the jacket as originally made is not an element of gain, and for that reason their later jackets are placed only at the upper end of the vertical cylinder. If heat is removed at all by the jacket there should be a gain. It may have been that the lower part of the jacket warmed the incoming gas and cut down the weight taken in, but as stated above, if the jacket removes any heat it should be an element of economy.

All parts of this cylinder and piston are easily accessible. This, together with the facts that there is no bottom wear or

friction on the cylinder and stuffing-box, from the piston and rod, in the vertical position, and that there is no stuffing-box exposed to high pressure, has led to the selection of a vertical single-acting compressor. The stuffing-box does not require such tight packing and this reduces the friction. In a double-acting compressor it would be difficult to use safety heads, and so the clearance must be greater. Of course this does not increase the work of compression except for friction, but it does cut down the volumetric efficiency, requiring a slightly larger cylinder. The use of two cylinders, whether of double or of single action, is advantageous in that if one must be disconnected for repair, the other may be operated alone and the plant kept in operation.

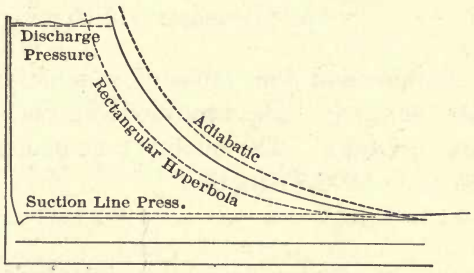


FIG. 37.—Indicator Card from Compressor with Guide Lines.

Fig. 37 shows an indicator card taken from such a compressor. If the clearance line, absolute zero line, suction-pressure line and discharge-pressure line are drawn on this card and then the adiabatic is constructed as shown by the method below, certain information may be had in regard to the operation of the compressor in addition to the knowledge of the power taken. The horse-power is worked out in the usual manner.

If the suction-pressure line is much above the back-pressure line, there is excessive valve friction due to the spring being too tight or the valve sticking. If the discharge pressure is much below the upper pressure of the card, the same may be said of the discharge valve. If the adiabatic falls below the compression line, there must be a leaky discharge valve, while a compression line below the adiabatic would mean a leaky suction

valve or piston. The jacket does remove heat and causes the compression line to fall below the adiabatic line in theory, but this amount is so small that it can hardly be noticed on the card. Hence, when there is a decided drop below the adiabatic, which is what is desired in theory, one must look for a leaky piston or suction valve, as the ordinary jacket could not produce the result.

The construction of the adiabatic is one which would have to be made by use of an equation of the form

$$pv^n = \text{const.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

If the compression is dry, the adiabatic is of the form

$$pv^{1.33} = \text{const.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

For wet compression the value of n must be computed for any given condition. The conditions at the ends of compression must be known. The quality x at one end is related to that at the other by the equation

$$s'_1 + \frac{x_1 r_1}{T_1} = s'_2 + \frac{x_2 r_2}{T_2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

s' = entropy of the liquid;

$\frac{r}{T}$ = entropy of vaporization;

x = quality.

In this x_1 may be found from x_2 since the two pressures are known, or as is usually the case, x_2 is made unity and then

$$s'_1 + \frac{x_1 r_1}{T_1} = s'_2 + \frac{r_2}{T_2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Having x_1 and x_2 the volumes per pound may be found by

$$v_1 = x_1 v''_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$v_2 = x_2 v''_2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

v = vol. of 1 lb. of mixture;

v'' = vol. of 1 lb. of dry vapor.

Then n is given by

$$n = \frac{\log \frac{p_2}{p_1}}{\log \frac{v_1}{v_2}} \quad \ldots\ldots\ldots \quad (7)$$

After the n for wet compression is found, the equation is known. Having the values of n, the volume v_1 from the zero volume line and the pressure p_1 from the zero pressure line are measured in inches and then by assuming other volumes the pressures at those points may be found by

$$\log p_x = \frac{\log p_1 + n \log v_1}{n \log v_x} \quad \ldots\ldots \quad (8)$$

These are tabulated for $n = 1.33$ for the card.

Point	1	a	b	c	
Volume in inches	2.16	1.20	0.80	0.70	
Pressure in inches	0.24	0.53	0.90	1.13	

The motor driving the compressor may be one of various types. For large compressors efficient Corliss engines are used for refrigeration, although for ice-making where distilled water is needed, less efficient engines are used. Gas engines are used at times and electric motors are of great value for small plants in hotels, hospitals, stores and residences.

One of the early successful ammonia compressors, used in the days of the introduction of mechanical refrigeration, which has remained one of the leading compressors, is that built by the de La Vergne Machine Company. The cylinder of their vertical type is shown in Fig. 38. This is their vertical double-acting compressor. The head of the cylinder contains several

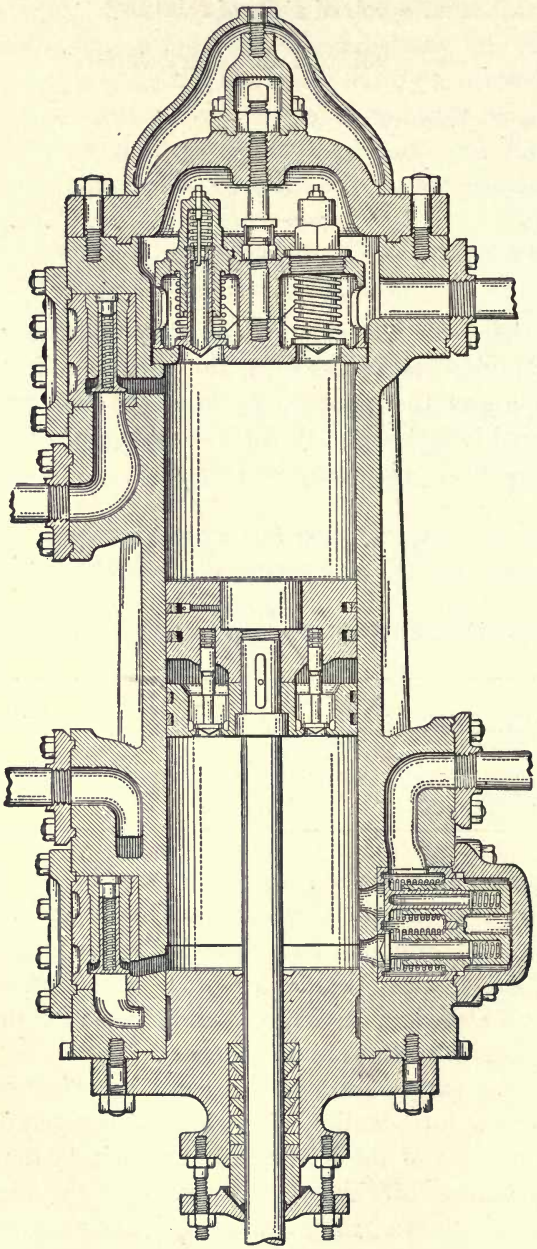


FIG. 38.—De La Vergne Vertical Double-acting Cylinder.

delivery valves placed in a casing. The suction valves are placed in casings inserted in the sides of the cylinder. Each valve is held to the seat by means of a spring and is arranged to be guided by a long sleeve around a central spindle. This forms a dash-pot action and prevents slamming. The suction valves are in cages forced into radial recesses in the cylinder. By removing the cover of the recess, the valves and their seats may be removed for examination or repair since the valve cages include valves, seats, springs, and dash pots. The head discharge valves are placed in a casing or housing. This is held against a projecting part of the cylinder casting making a gas-tight joint by the head pressure in addition to the pressure from a set screw attached to the main head of the cylinder. This set screw has a jamb nut on it and to care for the ammonia leakage around the threads, a cap is fastened over the top. This cap and the main cylinder head are made gas tight by lead-ring gaskets in a groove into which a projecting ring fits. These rings and the method of holding the head are made clear in the picture. The set screw holding down the valve housing is in reality a safety device, for should scale or other obstruction fall on top of the piston the bolt would break when the obstruction was brought up against the valve housing at the top of the stroke. The lower discharge valves are attached to housings at the bottom of the cylinder. The stuffing-box gland is shown in the figure and owing to the peculiar use of oil in this cylinder there is no provision for introducing oil into the gland.

A peculiar feature of this compressor and that of the de La Vergne Company is the introduction of a spray of light paraffine oil on each stroke. After spraying this oil forms a thin layer over the top of the piston on the down-stroke and one on the lower cylinder head on the up-stroke. In this way the piston and piston rod are sealed with oil, thus cutting down the tendency to leak. This oil also fills up the clearance space at the top end of the stroke. The excess oil is driven out through the valves. This of course reduces the clearance to zero. At the lower end of the stroke the oil would not flow away readily, so valves are introduced into the piston allowing oil to enter the hollow part

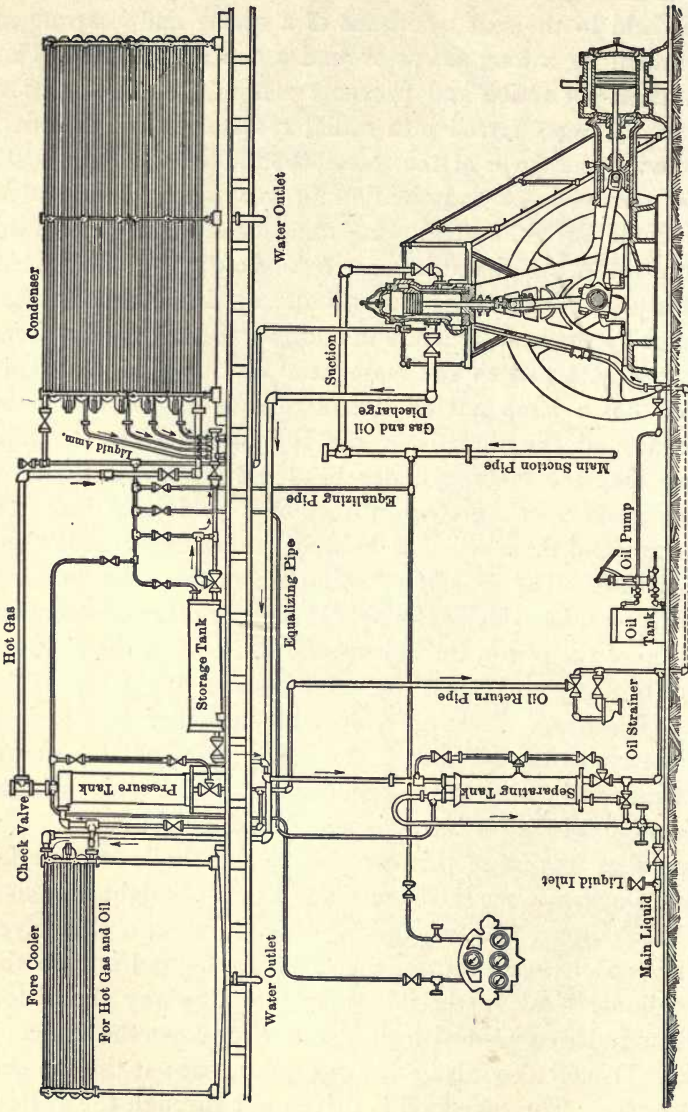


FIG. 39.—Diagram of a De La Vergne Refrigerating Plant.

of the piston. From this space the oil discharges through the upper discharge valve, when it is connected with this space at the lower end of the stroke by an opening in the side of the piston. In this way the oil is carried out without the danger of breaking the compressor. This injection of liquid also absorbs some of the heat of compression and makes the work less.

The piston is fairly deep considering the fact that the oil seal cuts down the amount of leakage. This also reduces the friction. The piston rod is held to the piston by a circular nut and projecting collar. The construction with two parts is clearly shown.

The false cap held on the cylinder head by the tap bolt is for finish only.

The suction and discharge pipes are attached by means of flange unions.

The use of oil requires additional apparatus to recover the oil from the discharge. The heated gas is first passed through the fore cooler, Fig. 39, and after being cooled it is taken to the pressure tank, where the oil separates out and the remaining gas goes to the condenser. The oil taken out goes back through the strainer to the engine; other oil which enters the condenser is finally separated from the liquid ammonia in the separating tank. The oil here separates and passes over to the compressor, being sucked in on the proper stroke. Fresh oil may be added to the system by the oil pump when needed.

The liquid ammonia is taken from the storage and separating tanks by the main liquid line to the various expansion coils in rooms or brine tank. The suction is brought back to the main suction pipe of the compressor.

Fig. 40 illustrates a horizontal type of compressor brought out by this company. The suction valve *A* opens into the passage *B*, which is connected to the cylinder *C*. The suction valve with its seat, spring and dash pot are in a housing which is held in place by a bonnet or cover-plate. The discharge valve *D* is in a similar housing. Either of the valves may be examined by simply removing the head. The housings are

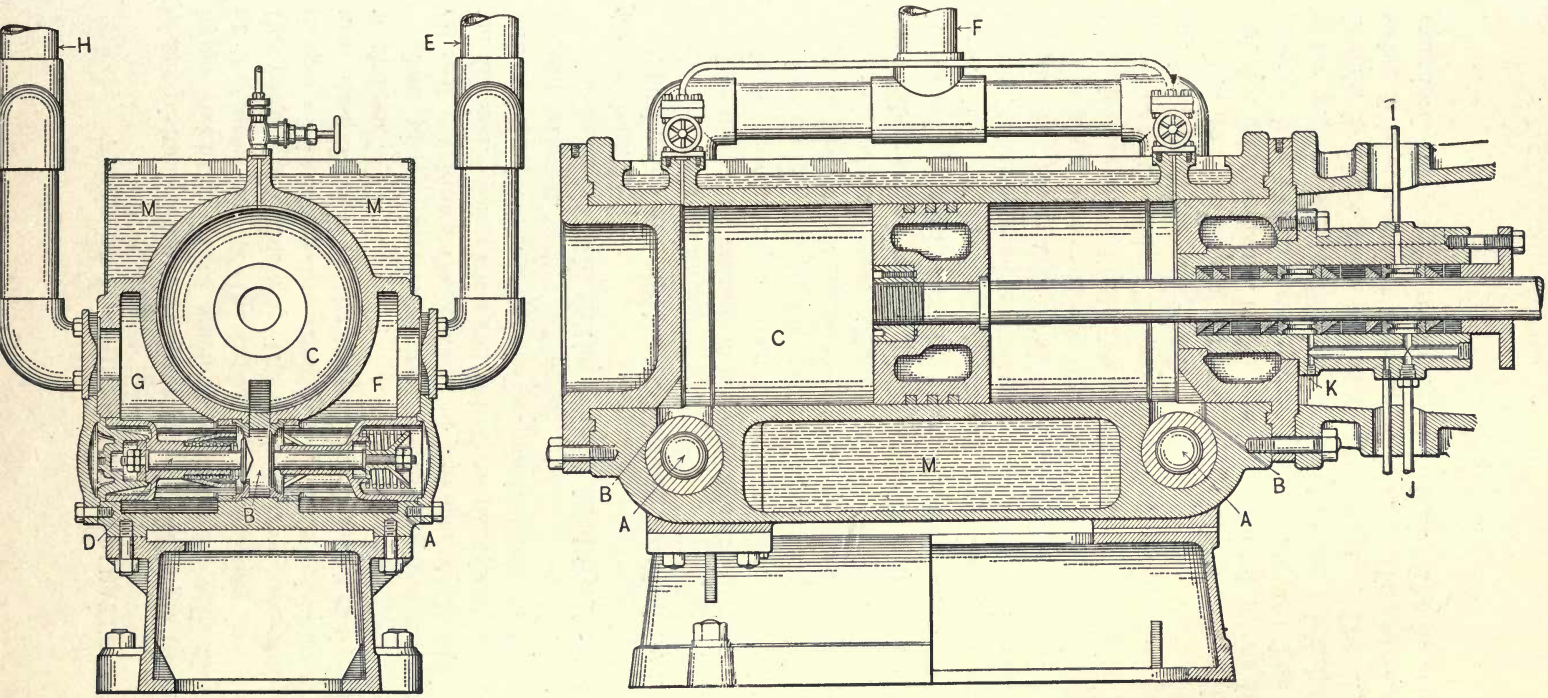


FIG. 40.—De La Vergne Horizontal Compressor.

arranged with slots so that gas from the suction main *E* enters the space *F* and goes through the valve from this into the cylinder. In the same way the discharge from the valve *D* passes through *G* into the discharge main *H*. In this cylinder there is no chance for the valves falling into the cylinder and any scale or obstruction would tend to fall into the space at the lowest point of the cylinder barrel. The piston and its attachment to the rod are clearly shown. The double-lanterned stuffing-box is shown. This is due to the fact that there is no oil lying around the rod as in the former case. The lubricating oil enters at *I* and is taken out at *J*. At *K* in certain cases, a connection is made to the suction pipe to remove any ammonia which has leaked past the first set of packing rings.

The cylinder is surrounded by the jacket *M*.

The cylinder of a horizontal double-acting compressor of the Frick Co. is shown in Fig. 41. This company builds vertical compressors which are very similar in general features to the compressor of the York Manufacturing Co., so that no section of that type will be shown. The cylinder is provided with valves in the spherical heads arranged in radial lines. They are arranged in this manner to increase the valve area for a given diameter of cylinder while using a small amount of clearance. There are usually two suction valves and two discharge valves on each end. The two upper valve boxes are connected, as are the two lower discharge boxes. As shown in the figure, these valves are so arranged that the seats, springs and valves may be removed with the valve housings by simply removing the bonnets. The long stuffing-box, the wide piston, the packing ring for the lead gasket in the heads and bonnets, the water jacket and the other peculiar features of ammonia compressors are clearly seen. The piston rod is properly attached by a nut. Some builders attach the rod by peening over the end after forcing the rod on the piston. This is not good practice and should not be resorted to unless absolutely necessary. It is better to use some form of nut or cotter pin. The piston is made in two parts, making a simple core arrangement in casting. To stiffen the cylinder, long

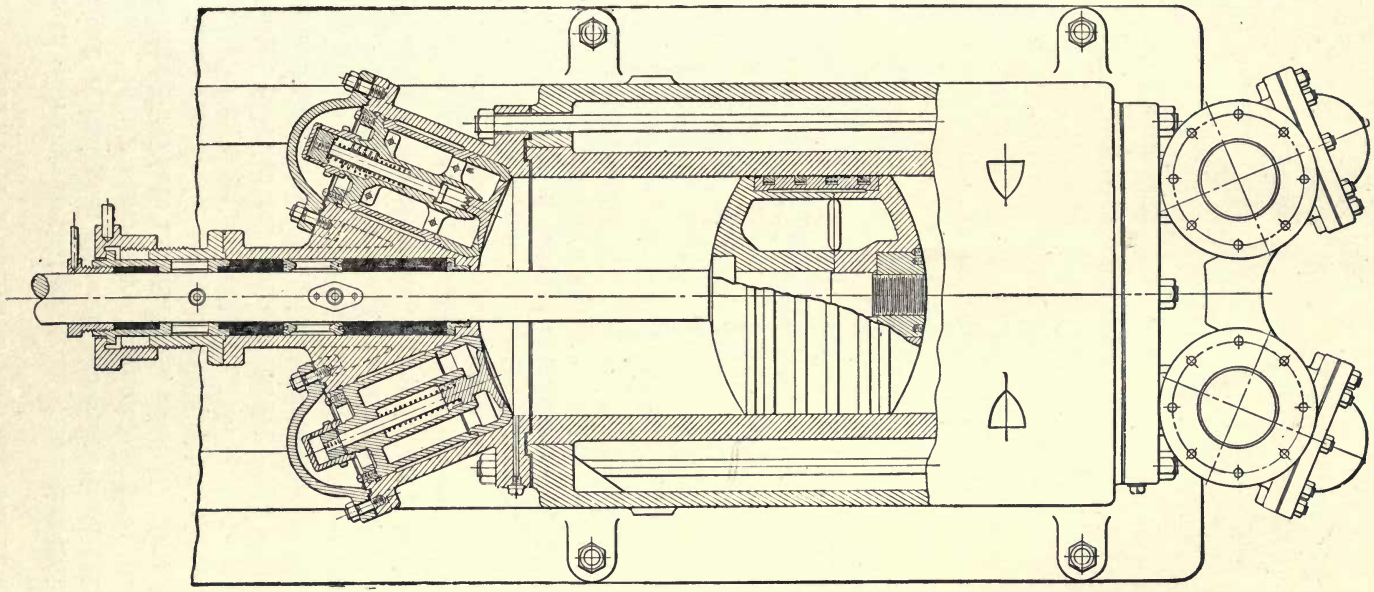


FIG. 41.—Section of Frick Horizontal Compressor.

through bolts are passed from one end to the other, thus relieving the cylinder of strain. The stuffing-box is provided for an oil-supply attachment, and an ammonia pipe returns to the suction pipe gases which leak out; these features are shown by dotted circles at the center of the rods.

For the proper operation of compressors it is necessary at times to remove the vapor from the cylinder. To do this there

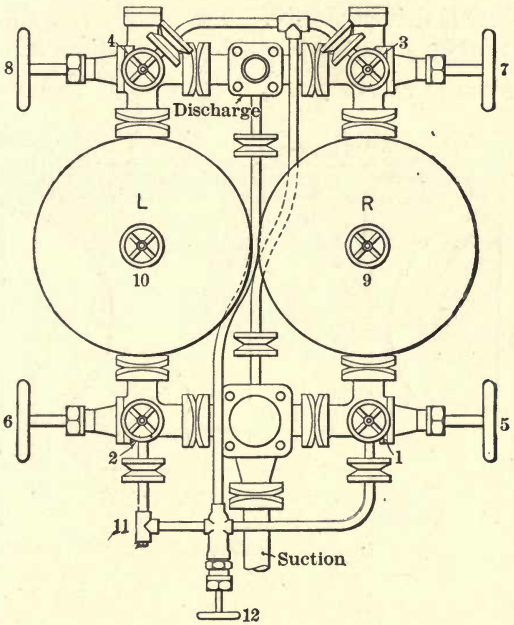


FIG. 42.—Frick Manipulating Valves for Small Compressors.

are certain by-pass arrangements common to most compressors. The arrangements used by the Frick Company are shown in Figs. 42 and 43. To exhaust the vapor from one compressor the machine is shut down and all of the valves are closed. The purge valve 10 is opened. This permits gas to escape. The machine is now operated slowly. The cylinder head of *L* may be removed. To exhaust *R*, the same method is used.

To exhaust the condenser and store the ammonia in the expansion coils, close all valves after shutting down, then open

the valves 1, 2, 3, 4 and 12 and start the machine slowly. The gas for compressors is sucked through valves 1, 12 and 2 from the discharge main, while after compression it is discharged through 3 and 4 into the suction piping.

To admit air into the high side for testing, close the suction valves 5 and 6, and leave the discharge valves 7 and 8 open. Open valves 1 and 2, removing the plug from tee 11. Valves 3, 4 and 12 are closed. Air is then drawn in by 1 and 2, when the compressor is driven slowly.

To admit air for testing low side, the suction valves 5 and 6 and the discharge valves 7 and 8 are closed. Valves 1, 2, 3, 4

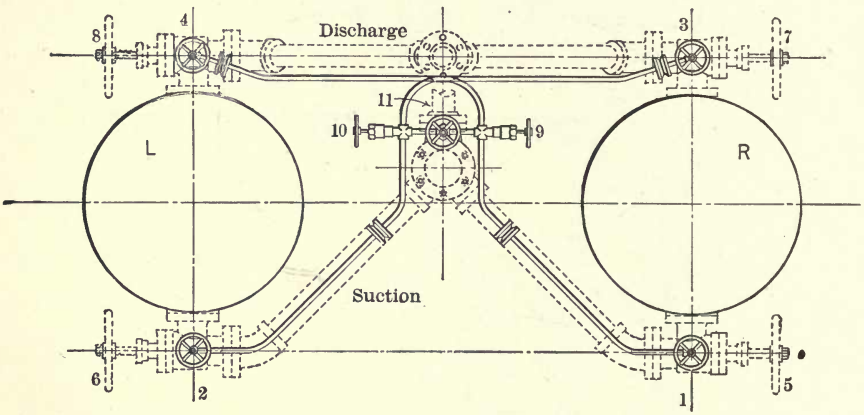


FIG. 43.—Frick Manipulating Valves for Large Compressors.

are opened and the plug in tee 11 is removed. Air will enter 1 and 2 when the compressor is run slowly and the discharge is passed into the suction main by 3 and 4.

Before air is introduced on either side the ammonia is exhausted from that side by pumping the ammonia into the other until a low vacuum exists on the side from which the vapor is being pumped.

For larger compressors a different arrangement of pipes is used, as shown in Fig. 43. In this case, valves 1 and 2 are attached to valves 5 and 6 and then valves 9 and 10 are added to the pipes running from 4 to 1 and from 3 to 2, while valve 11 connects to the suction pipe.

To exhaust the compressor *R*, all valves are closed after shutting down the compressor. Then stop valve 8 is opened with 2 and 3. The valve 10 when closed prevents any connection to the suction at that point. If the compressor is started slowly, the compressor *L* draws from *R* and frees it of ammonia. Valve 2 opens inside of 6.

To exhaust the condenser the valves opened are 8, 1, 4, 3, 10 and 11. In this case the gas is sucked from the discharge pipe through 8, 4, and 1 into *R* and then the compressor vapor passes over through 3, 10 and 11 into the suction mains. The

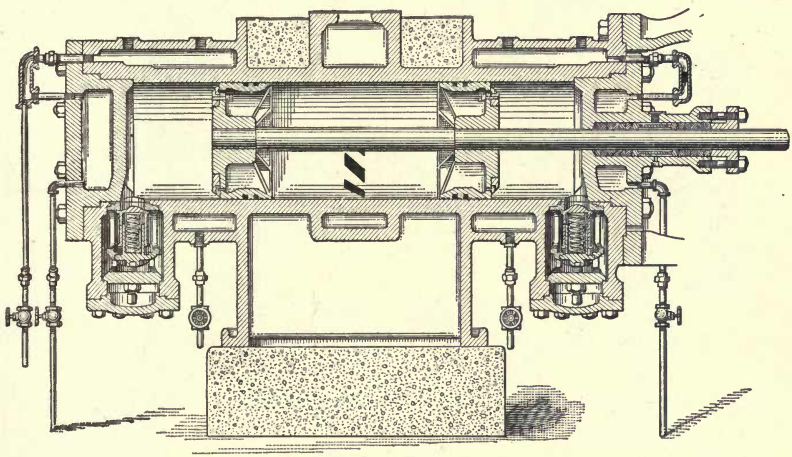


FIG. 44.—Section of Arctic Horizontal Double Single-acting Compressor.

other compressor could have been used. To empty the suction the compressor is operated in the usual way with the reducing pressure valve closed.

One recent improvement in the arrangement of compressors is that of the Arctic Machine Co. in their center inlet horizontal double-acting compressor shown in Figs. 44 and 45. In this compressor the inlet valves in the piston are similar to the hurricane inlet valves of the Ingersoll-Rand air compressors. To avoid the piston-rod inlet connection, the two piston faces are separated a distance equal to the stroke of the compressor and the center of the piston barrel is cast with openings leading

from a ring passage into the cylinder bore. The suction vapor enters this passage and the space between the two piston faces. There are a series of openings extending through the piston face near the periphery and these are covered by a ring of metal. When the piston starts from one end, a vacuum forms behind the piston, and the gas between the piston faces, being at a higher pressure, forces the ring out against a stop and enters behind the piston with little change in pressure. At the other end of the stroke, the acceleration of the reciprocating parts when the piston begins the return stroke closes this valve and the piston begins to compress. In this way

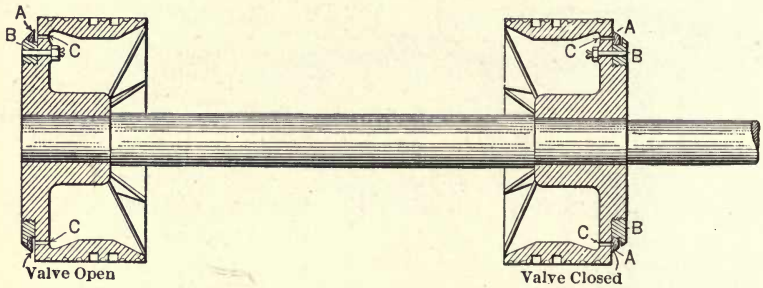


FIG. 45.—Suction Valves and Pistons of Arctic Compressor.

the compressor acts rapidly on the suction stroke with little drop in pressure.

The discharge valves and housing are connected to the lowest part of the cylinder at each end, thus caring for scale and liquid. These valves may be examined by removing the bonnet. The springs and dash pot are seen in the figure.

The cold ammonia coming to the center does not affect the stuffing-boxes and the jackets are removed from the cold parts of the cylinder. An insulating filling is placed at the center to cut down radiation. The heads are jacketed as well as the barrel ends. The path of the water is seen.

The stuffing-box is shown with its oil connections. There is also a connection leading from the stuffing-box to the suction main.

The piston detail is shown in Fig. 45. The valve disc is

the ring *A*, the motion of which is limited by the guard ring or projection *B*. The openings *C* are distributed around the periphery of the piston. The method of attaching the piston is proper in this case as the load is taken by a shoulder. Each of the piston faces is in reality a single-acting compressor. The action of the suction valve is so free that, according to reports on the compressor, the suction drop was only 1 lb. at 300 R.P.M.

The valves of the compressor of the Triumph Ice Machine

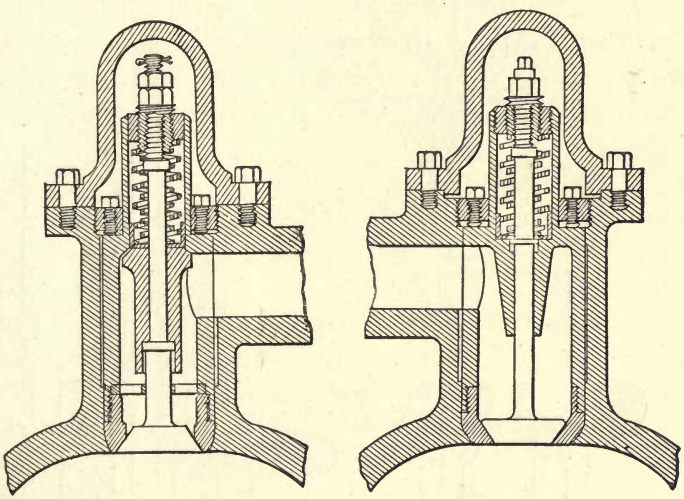


FIG. 46.—Valves of the Triumph Ice Machine Co. Compressor.

Co. are shown in Fig. 46. These valves and their seats are held in a housing or cage which is held in place by a nut screwed into the valve cavity and containing set screws to hold the cage tight against the head casting. The suction valve spindle is held up by two springs each with a separate adjusting collar. These collars may be held tight by jamb nuts or set screws. The small collar at the lower end of the suction spindle acts as a dash pot.

In the discharge valve there are two springs holding the valve down and a cup acting on a shoulder or collar on the

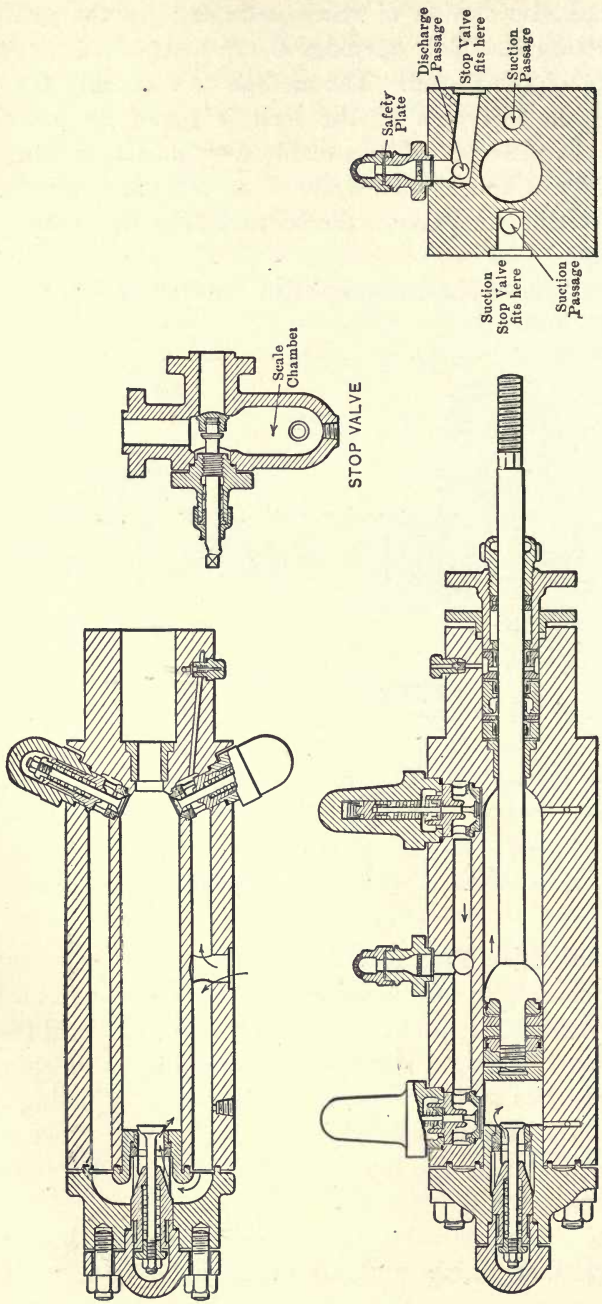


FIG. 47.—Cylinder of Kroeschell CO_2 Compressor.

spindle serves as a piston and dash pot. In each case the long spindle guide keeps the valves in line.

The cylinder of a carbonic acid machine will be shown because of the special features due to the high pressures carried. One of the best known compressors of this type is that of Kroeschell & Co. and is shown in Fig. 47. The cylinder is a rectangular steel forging in which the ports are drilled from one end. The various valves are placed in housings which are held in place by bonnets. These valves are operated by springs as shown. The suction valves have helical springs while a cap on the top of the spindle limits its movement. The discharge valves have a spring to support them and a flat spiral spring to force them back. The seats are held in by bonnets or cages and may be removed readily. The bonnets are made tight by fiber washers.

The piston is packed with cup leathers to sustain the high pressure. One end is flat because of the head and the other end is spherical in order to place the two suction valves and one discharge valve.

The piston rod is packed with a series of cup leathers with provision for circulating oil.

The two suction passages are connected around the head valve, the suction pipe entering one of the passages at the center as shown. The discharge passage is connected to the top face of the cylinder. Each of these is controlled by a valve. The valves, Fig. 47, have drips or scale chambers at the bottom to catch dirt. The discharge passage is connected to an opening covered by a thin plate which will break when excessive pressure is brought on the discharge by shutting off the discharge stop valve or by a stoppage of the system. A relief valve is also fixed on the suction side to relieve excess pressure.

The parts of this compressor are made of steel, due to the heavy pressures. The remaining part of the compressor is similar to any other type.

Fig. 48 shows one of the Kroeschell marine compressors in which the **double-pipe brine cooler** and **double-pipe condenser**

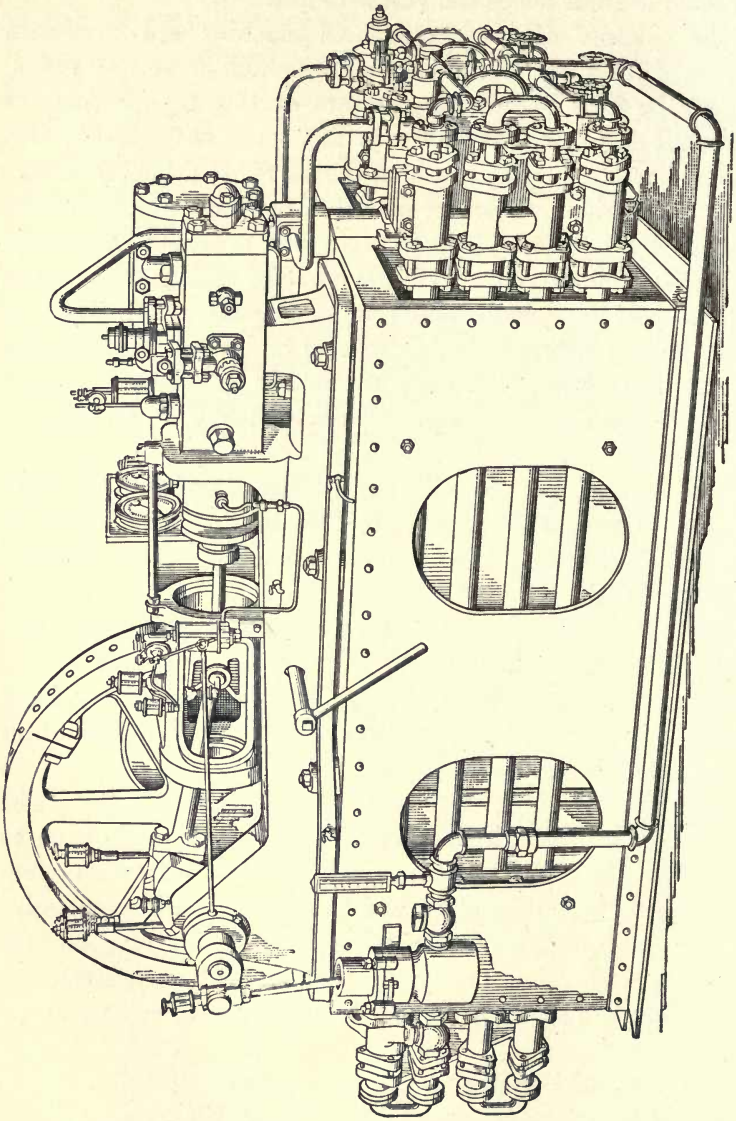


FIG. 48.—Kroeschell Compressor, Condenser, and Cooler for Marine Work.

are placed in the base of steel. The steam engine driving this is seen at one side of the compressor.

One of the latest types of machines in which the compressor, condenser, brine cooler, and pipe system are contained within the same casing is shown in Figs. 49 and 50. This is the Audiffren-Singrun Refrigerating Machine, as sold by the H. W. Johns-Manville Co. In the spherical case *A* is a hollow shaft *B*, supporting as an axis a casting *V* which is so heavily weighted by *W* that it will not turn. This casting carries the trunnions *TT* of a cylinder *C*, the piston of which is connected to a rod attached to the strap of an eccentric sleeve *D* on the shaft. If the whole casing is turned and with it the shaft, the heavy weight remains down and the piston in the cylinder is drawn in and out by the eccentric while the cylinder oscillates. Thus oscillation of the cylinder between the faces of the suspended casting causes the face of the cylinder casting to oscillate over the face of the right-hand casting which contains holes. In this way the ports of the two ends of the cylinder are connected to suction ports *N* in the hanging casting at the proper time in the same way as the distribution of steam is accomplished in the oscillating engine. In this way SO_2 vapor is admitted to the cylinder from the annular space *E* between the two shafts and the space *F* when the holes at *G* in the cylinder and face come opposite. The vapor is compressed in the cylinder and when the proper pressure is reached the discharge valves at *H* open and discharge the vapor into the casing *A*. The casing revolves in a tank *I*, Fig. 50, containing the cooling water. This condenses the sulphur dioxide and the liquid collects at the outer part of the casing *A*, and is caught up by a scoop *M*, and is conducted to the reservoir *J*. It is then delivered to a regulating float throttle valve after the lubricating oil is removed from the SO_2. The oil flows over at *U* into the chamber *O* in which the cylinder is placed. Thus the eccentric and cylinder are flooded with oil. This whole region is under pressure, so that there is no leakage from the compressor. There is a tendency for the oil to enter around the piston rod and between the valve faces. The spring *X* holds the system against the sliding face. The

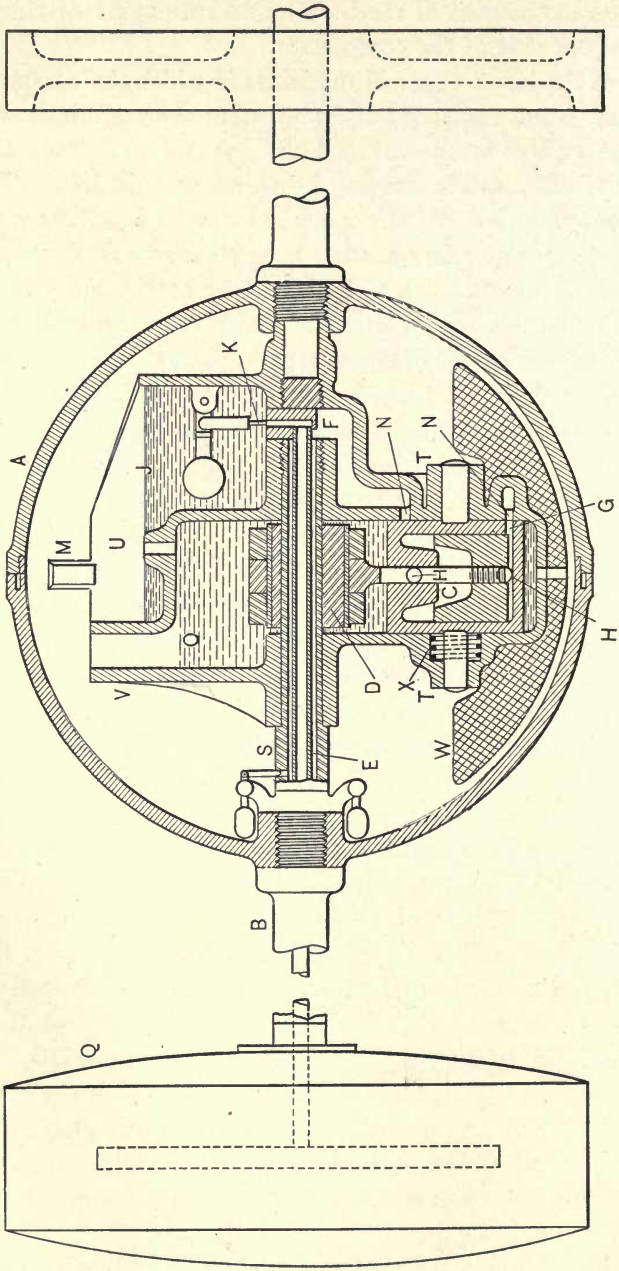


FIG. 49.—Audiffren Singrun Machine.

liquid SO_2 at low pressure after passing the throttle valve travels along the inner pipe extending between the two vessels and finally settles to the circumference of the other spherical vessel, due to centrifugal force, and it is evaporated as it removes heat from the brine in the tank *R*, Fig. 50. The vapor is returned through a space formed between the two pipes between *A* and *Q*, and passes into the SO_2 compressor. The complete system is contained within a tight set of vessels and pipes, and there are no moving joints to be kept tight. There is no danger of leakage. An extension on the right-hand vessel serves as one journal for the system and the intermediate pipe serves

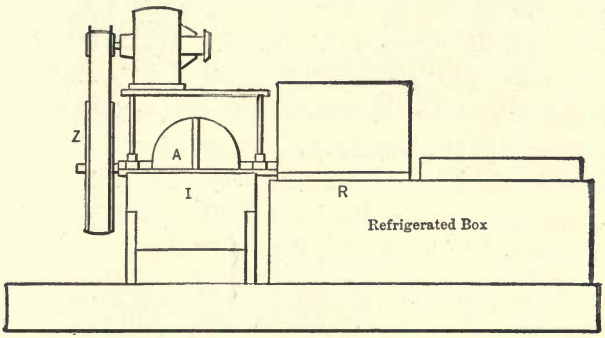


FIG. 50.—Audiffren Singrun Apparatus.

as the other. There is little weight on the journals, as the buoyancy from the immersed vessels supports much of the weight. The gas pressure in *A* tends to hold the oscillating piston against its face in addition to the spring pressure, and keeps the sliding joint tight. Should the condensing water be shut off and the temperature rise, the high pressure developed would finally be sufficient to cause the weight to rotate and so prevent a further rise in pressure. The small valve at *S* is held down, when the apparatus is in operation, by centrifugal force, but upon stopping the machine this valve is opened by the weight of the balls, thus equalizing the pressure. The following table gives the data for these machines:

Size of Machine.	Capacity in Tons.		Power Required.	R.P.M.
	Refrigeration.	Ice.		
2	0.19	0.13	0.4 to 0.6 H.P.	380
3	0.48	0,32	1 to 1.5	280
4	0.96	0.66	2 to 2.25	190
6	1.92	1.32	4 to 4.50	140

The great advantage of such a machine lies in the fact that there is no manipulation of valves, stuffing-boxes, gauges or oiling devices.

In Fig. 50 the general arrangement of this apparatus is seen. The cooling water in *I* liquefies the SO_2, while the evaporation of the SO_2 cools the brine in the tank *R*. If this brine is cooled completely there will be no evaporation of SO_2, and there will be no gas sent back to the compressor, and consequently none will be liquefied in the case in *A*. Hence, after a short time the level of the liquid in *J* will be such that the float valve *K* will be closed off and no more liquid SO_2 can pass over to *R*. The motor is attached to the pulley *Z*.

The above gives the necessary details of the compression machines. The parts of the absorption machine will now be considered.

Generators. In Fig. 19 a cross-section through a generator is shown. This is seen to be made of a circular tank with flanged ends and dished heads, containing several coils of steam pipes connected to manifolds. This is of cast iron, and bolted to a T-connection is the analyzer, containing a series of trays over which the liquor from the exchanger and rectifier flows. The pipes are so arranged that the gases have to pass through the liquid. The cast head, dished to give strength, is shown.

In Fig. 51, the generator of the Henry Vogt Company is shown. In this strong liquor flows into the top section of the generator from the exchanger. The excess liquor flows from this at the left-hand end to the next section, and the excess from this might flow into a lower section if used. In this way the liquor travels the whole length of the section before leaving it.

The weak liquor is taken off at the right-hand end of the lower section. The connections are made to give a definite liquid level in each section. The vapor formed in each section is taken up through a main pipe to the rectifier. The steam coils are connected to manifolds which are connected together

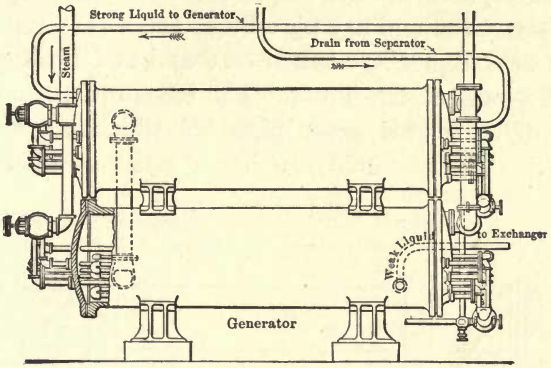


FIG. 51.—Vogt Double Cylinder Generator.

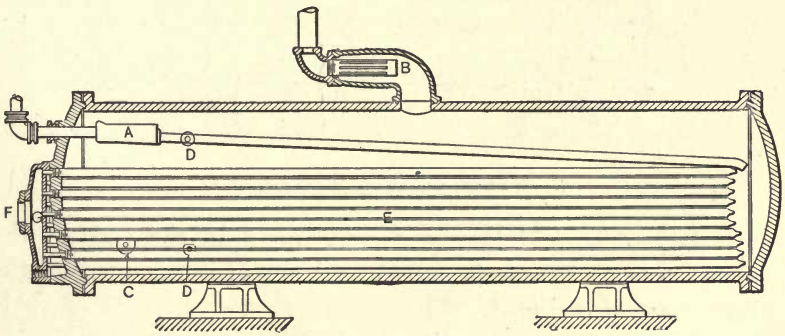


FIG. 52.—Vogt Modern Generator.

on the steam side and at the discharger end. The cylinders and heads are made of cast iron and the supports are made to carry these from the lower sections.

The analyzer is arranged at times to cause the vapors to travel up through the strong liquor which flows over perforated plates. The gas is then carried to the rectifier.

Fig. 52 is a section of one of the later forms of Vogt gen-

erator. The generator is made of semi steel pipe with dished heads using a tongue and groove packing. The strong liquor pipe passes through a stuffing box and is attached to the analyzer header *A*. From this point the liquor is carried into a number of tubes forming the analyzer in this case. These are in the space through which the heated gases pass on the way to the outlet and dry pipe *B*.

The weak liquor is taken from *C* and at *DD* the glass gauge gives the level of the liquor. The evaporation occurs at the surface of the closed steam pipes, *E*, which are screwed into the head. The manifold cap has a series of small tubes, *G*,

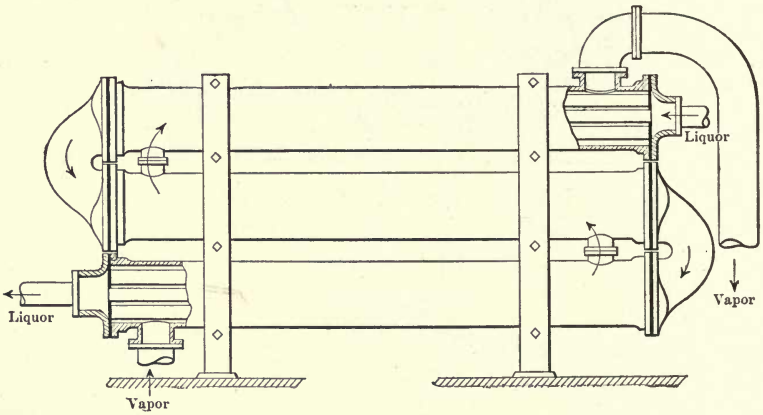


FIG. 53.—Vogt Rectifier.

attached to it, and taking steam from *F* to the ends of the closed tubes. Then the condensed steam is removed at the lower part of the head.

This construction is quite simple and effective.

Rectifier. The rectifier is made in several ways. In some cases, as in Fig. 19, it consists of a coil of pipe made up of return bends, through which the vapors flow to the separator and condenser while cooling water is passed over it. In other cases it is formed as a double-pipe condenser, the construction of which will be explained later. When this double-pipe apparatus is used, Vogt uses the strong liquor as cooling substance, passing it directly to the rectifier before it enters the exchanger.

Exchanger. The exchangers are of various forms. In Fig. 19 the form is a cast-iron cylinder with a coil within. This coil carries the weak liquor and while the strong liquor passes around this coil as it goes through the shell. The York Company and Vogt use a double pipe construction for this apparatus.

Weak Liquor Cooler. This is of double-pipe construction.

Absorber. The absorber is made in several forms. In all of them vapor enters at bottom and is distributed through a perforated pipe. The weak liquor is distributed near the top of the absorber, the flow of weak liquor being controlled by a

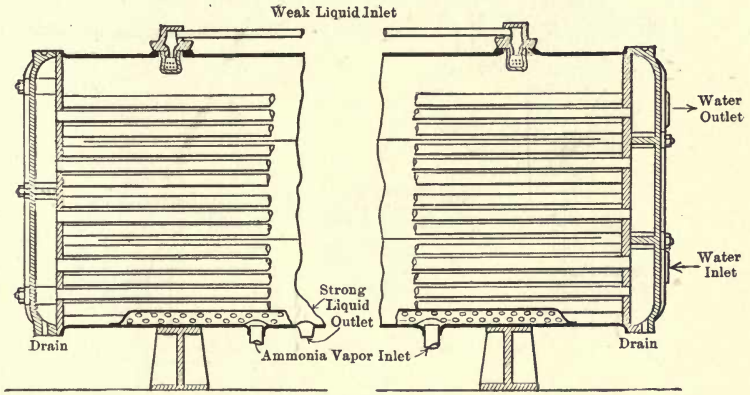


FIG. 54.—Vogt Absorber.

float shown in Fig. 55. This float is attached to the side of the a sorber. The action of the float is to control the admission of weak liquor by the valve *A*. The strong liquor is pumped from the bottom of the vessel, which is usually made cylindrical. In the absorber there are sets of tubes carrying cold water to remove the heat of absorption.

Fig. 19 gives the construction used by the Carbondale Co., while Fig. 54 is the absorber of Henry Vogt & Co. The drawing shows the construction and the manner in which there are four passes of the water.

The other parts of the apparatus being used with this and compression machines will now be described.

Piping. The piping for ammonia should be full weight or extra heavy wrought-iron pipe. The pipe is united with

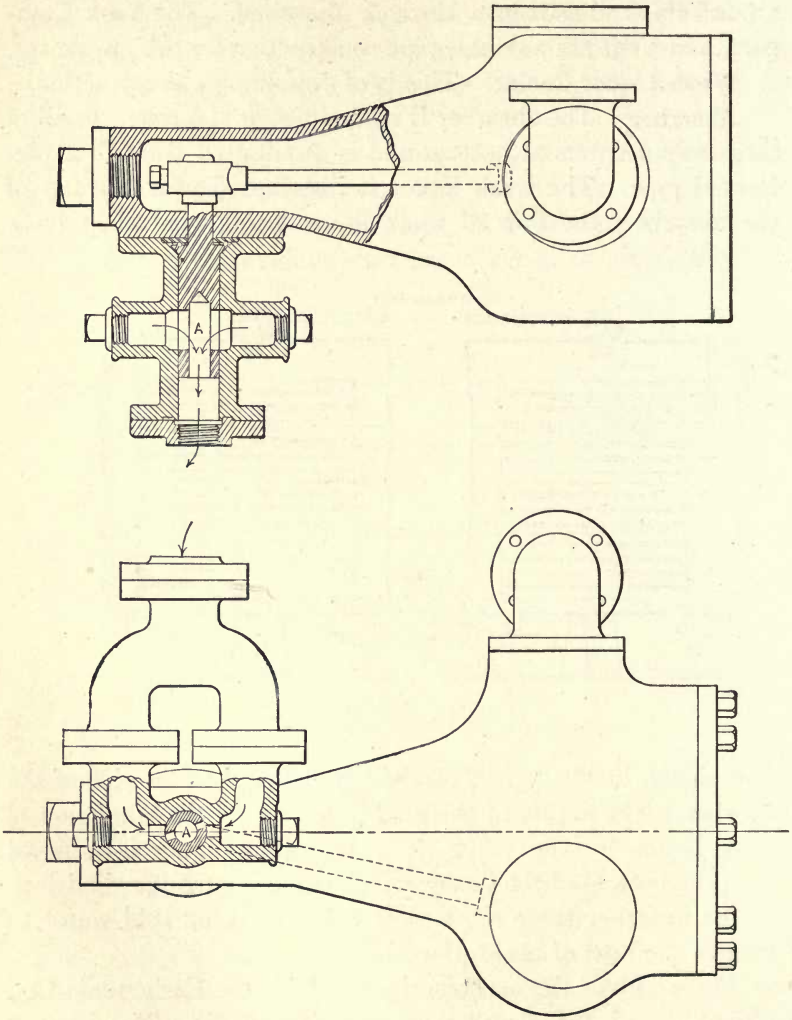


FIG. 55.—Vogt Regulator for Weak Liquor Inlet.

screwed fittings, flanged fittings and by welding. For use with CO_2 extra heavy pipe must be used. This is determined by the pressure to be carried and the opinion of the engineer.

Ammonia pressures will run as high as 200 lbs. per sq.in., while carbon dioxide may be 1000 to 1200 lbs. The best method of uniting these pipes is by welding, as there is no chance for leakage, although they cannot be dismantled easily. Welding is done by the use of thermit, the oxy-acetylene torch or by electricity. In this work the parts are clamped together tightly and thermit is ignited in a crucible, after which the aluminum oxide and the hot iron are poured into a mold around the pipe and produce a welding temperature. Thermit is a mixture of Fe_2O_3 and 2Al. It burns to 2Fe and Al_2O_3, producing a temperature so high that the molten iron and slag can heat the pipe to a proper point for welding. The thermit powder is held

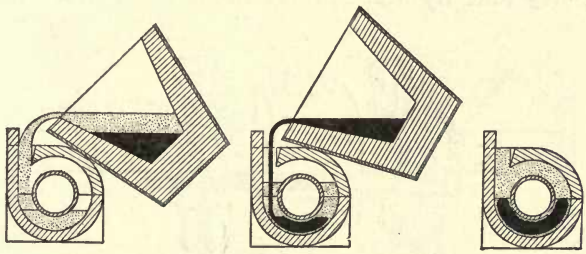


FIG. 56.—Thermit Pipe Welding.

in a crucible and after ignition is complete it is poured into the mold around the part to be welded.

Before welding the pipe ends are milled smooth by a special facing machine and are then clamped and held tightly together. An iron mold is then put around the pipe and the thermit poured in as shown in Fig. 56. When the operator feels the clamp, Fig. 57, yield he knows that the iron has reached a welding heat, and by pulling the clamps together and giving four quarter turns of the bolt the weld is made. After the weld is made the mold should be left ten or fifteen minutes if possible, and then removed. The iron and slag will fall away. Thermit is placed in bags with the proper amount for a given size pipe. In igniting it, it is customary to use about one-half of the package at first and then, after igniting it by means of an ignition powder and match, the remainder of the package is poured in. The

slag first forms a coating around the pipe, protecting it from the molten iron. The heat is used only to bring the iron to a welding temperature.

Tests of pipes welded in this manner have shown that the weld is as good as the pipe in tension, torsion and bending tests. The cost of thermit welding amounts to 75% of the cost of elbows or flanged joints for small sizes, while for four-inch flanged joints the thermit cost is greater than the cost of fittings.

The use of the oxy-acetylene torch is valuable in cutting as well as in welding. C_2H_2 is mixed with O_2 in the nozzle, and if just enough oxygen is introduced, the flame will consist of CO_2 and H_2O. The heat is produced by the breaking down of the C_2H_2 and by the formation of CO_2 and H_2O. An

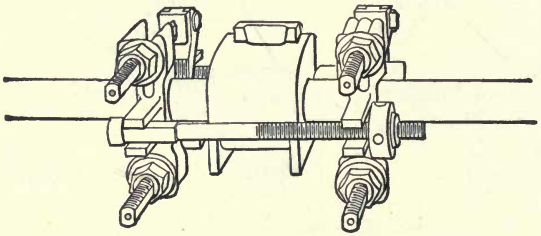
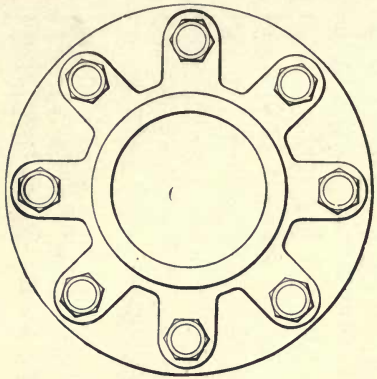


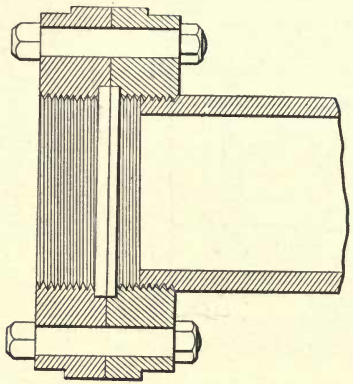
FIG. 57.—Thermit Welding Clamp and Mold.

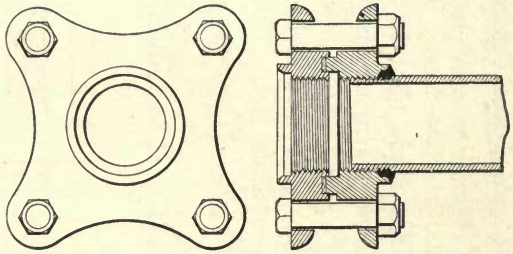
intense flame temperature is obtained. When welding is desired the mixture is as given above, and by pressing the parts together and melting a stick of steel by the flame to flow into the interstices a fixed weld is made. If it is desired to cut the metal, a correct burning mixture is used on the outer part of the flame and after this heats the metal, a flame rich in oxygen is thrown out from the center of the nozzle and burns a groove through the metal. This torch is particularly valuable for welding plate work. The temperature is so high that the H_2O is dissociated and the hydrogen burns on the outside of the flame.

In electric welding an alternating current of low voltage but great current strength is delivered from a transformer through large leads clamped to the pipes to be welded. The

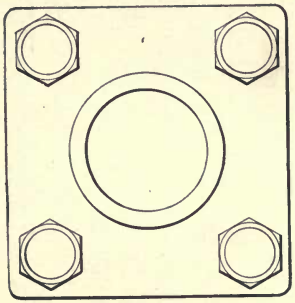


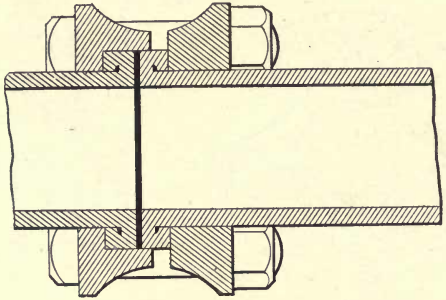
A. Frick Flat Flange.



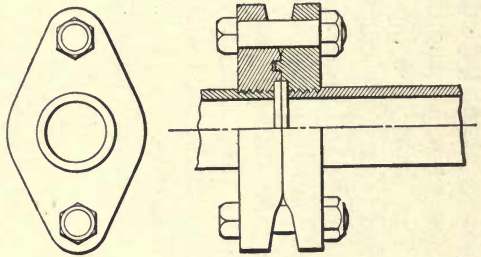


B. De La Vergne Grooved Flange.





C. Philadelphia Pipe Bending Co. Flange.



D. Frick Flange.

FIG. 58.—Flange Unions.

resistance at the butted ends to be welded soon causes these to become white hot and the metal is welded.

The fittings on pipes are usually screwed on. The taper thread is carefully cut and the joint made tight by a mixture of litharge and glycerine. This forms a cement and makes a good joint if the fittings are made tight. At times, with black iron, the threads are tinned over, when the solder makes a joint if they are screwed together when hot. This method should not be used with galvanized pipe. There are claims for each method.

The flanged unions, Fig. 58, are used for uniting sections

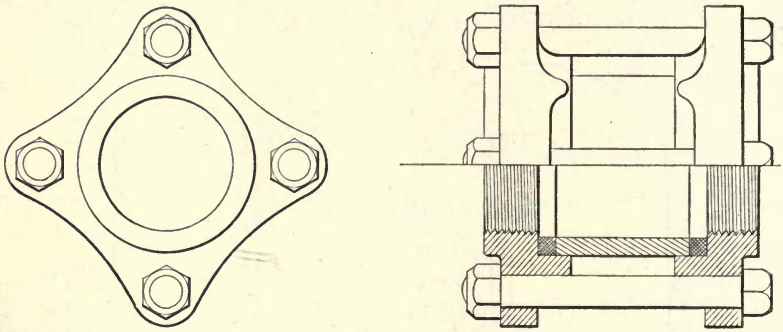


FIG. 59.—Boyle Union.

which may have to be separated. They are screwed to the pipe as shown in the figure or sometimes the type shown at *C* is used. *B* shows the flange joint of the de La Vergne Co. It has a cavity left at the upper end of the screwed portion of the flange into which solder may be left as it is forced out from the tinned threads when the flange and pipe are heated and forced together. This fills the space and threads at one end with solder so as to make a solid gas-tight joint. The flanges are made of a close-grained malleable iron combining strength and toughness, or else drop forgings or steel castings are employed.

The joint between the two flanges is made tight by a lead gasket which fits in a groove in one flange and is pressed down

by a projecting ring on the other flange. At times lead gaskets are placed between flanges as shown in *A* and *C*. Fig. 59

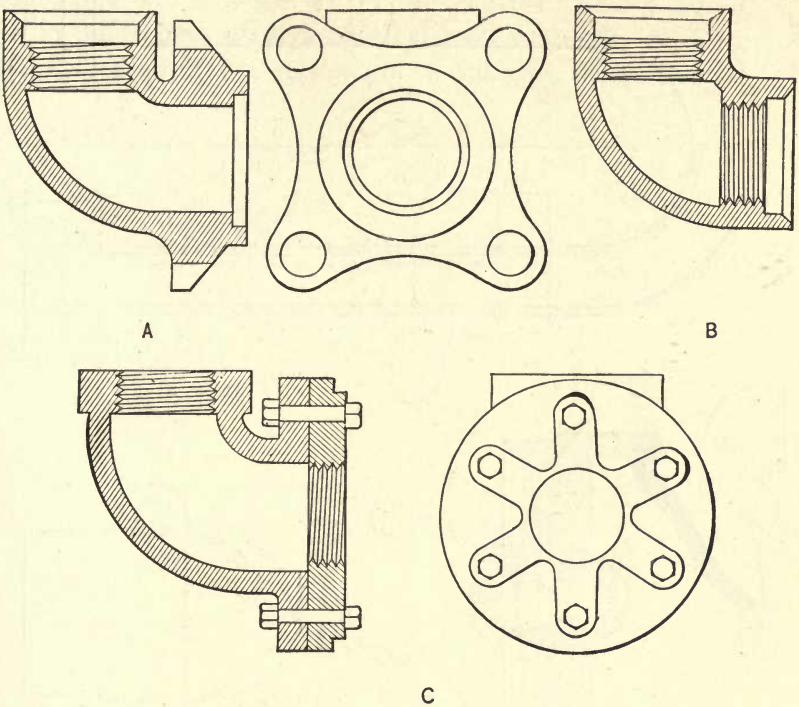


FIG. 60.—Elbows.

illustrates a Boyle union used in refrigerating work. In this a change of alignment is possible by properly finishing the ends of the nipple.

Where elbows are needed they may be screwed as shown in Fig. 60 *B*, and sometimes it may be necessary to use a flanged elbow, the flange being on one outlet as in the figure. Very often both ends of the elbow have flanges. A tee, Fig. 61, is used when it is desired to take off a side branch. Return bends are made as shown

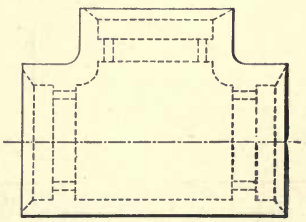


FIG. 61.—Tee.

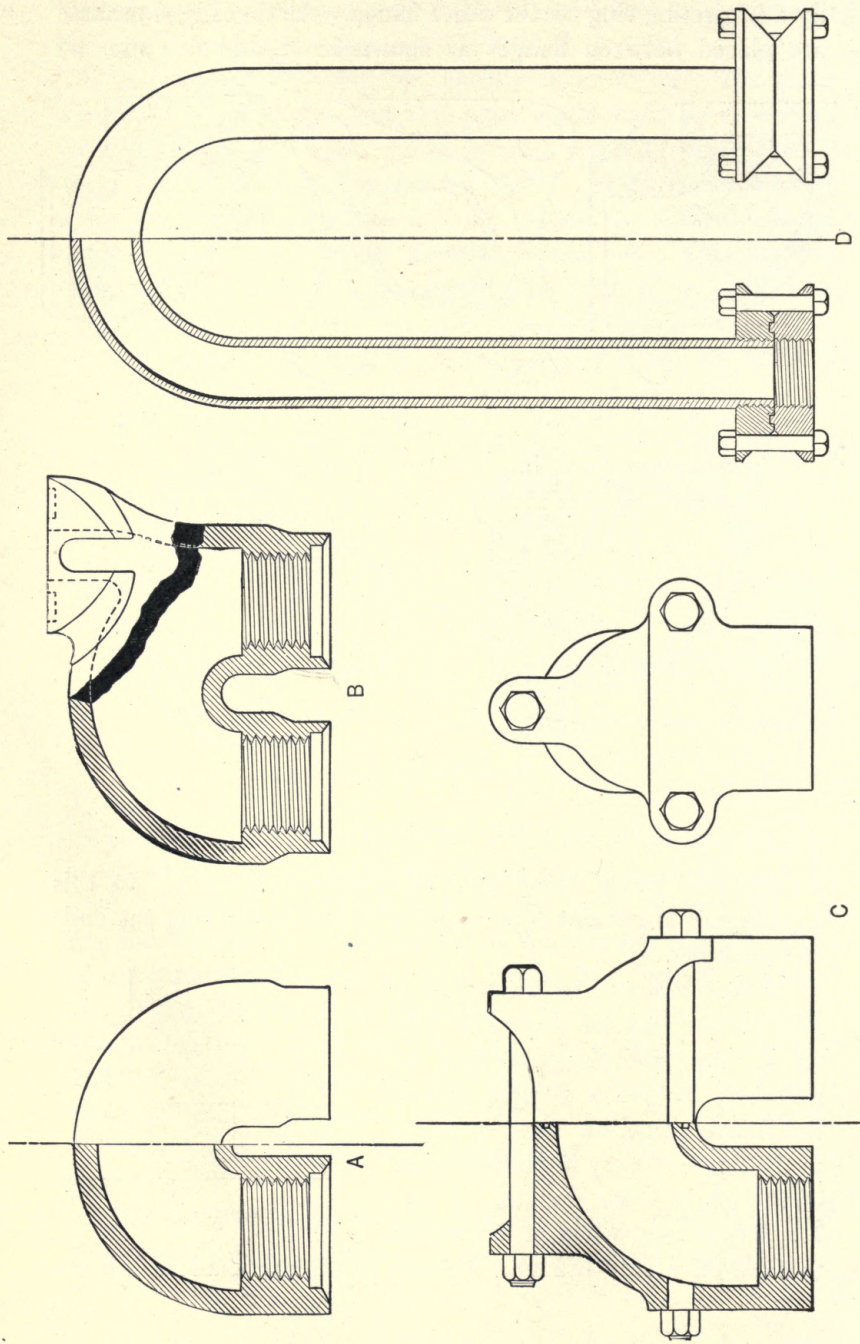


FIG. 62.—Return Bends.

in Fig. 62. These tees and bends show different arrangements used in ammonia work. Thus one bend *B* has an extra flanged outlet on it. It is a special return bend used on the de La Vergne condensers to carry off condensed ammonia. If it is desired to connect two sections of a coil, two flange fittings

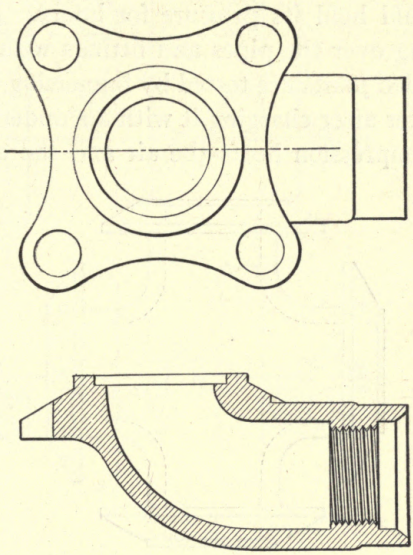


FIG. 63.—Flanged Return Bend.

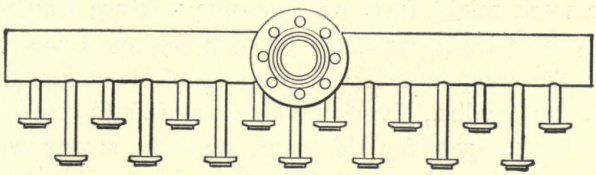


FIG. 64.—Branch Tee or Manifold.

known as return bend flanges are employed. This is shown in Fig. 63.

Manifolds, or headers, for the connections of a number of branches, are made by welding. They may take a number of special forms, depending on the peculiarity of design. Fig. 64 illustrates one with fifteen branches for the connection of the

different coils of a condenser. A cross, Fig. 65, is used at times when two lines are to intersect or three branches are to be taken from a line.

All of the fittings are extra heavy to allow for the high pressures, and after erection the whole system is filled with air under pressure. After closing the valves of the compressor the system should hold its pressure for hours. Leaks may be found by coating over the pipes and fittings with soapy water. In the shop welded joints are tested by immersing the apparatus in a tank of water after charging it with air under pressure.

Since the compression heats the air and the oil vapor from

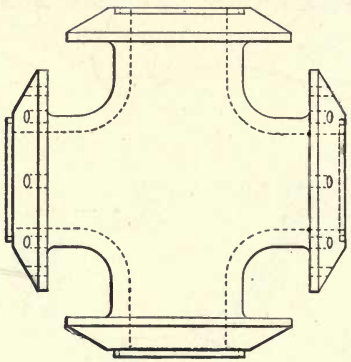


FIG. 65.—Cross.

the pipe work might form an explosive mixture which would ignite at the temperature due to compression, Block advises stopping the compressor for a while after reaching 50, 100, 150, 200, and 250 lbs., giving the air some time to cool.

The pipe hangers for this work must be strong and well supported, as many pipes are filled with brine and loaded on the outside with ice and snow. The weight of these must be added to the weight of pipe in figuring the strength of the hangers. Fig. 66 illustrates several methods of supporting the pipe.

The valves used as stop valves on the vapor line are of various forms with strong flanges and stuffing-boxes. Usually the seat has a soft lead ring for giving a tight joint, and the stuffing-box is long. The bonnet of the valve is bolted on the main body,

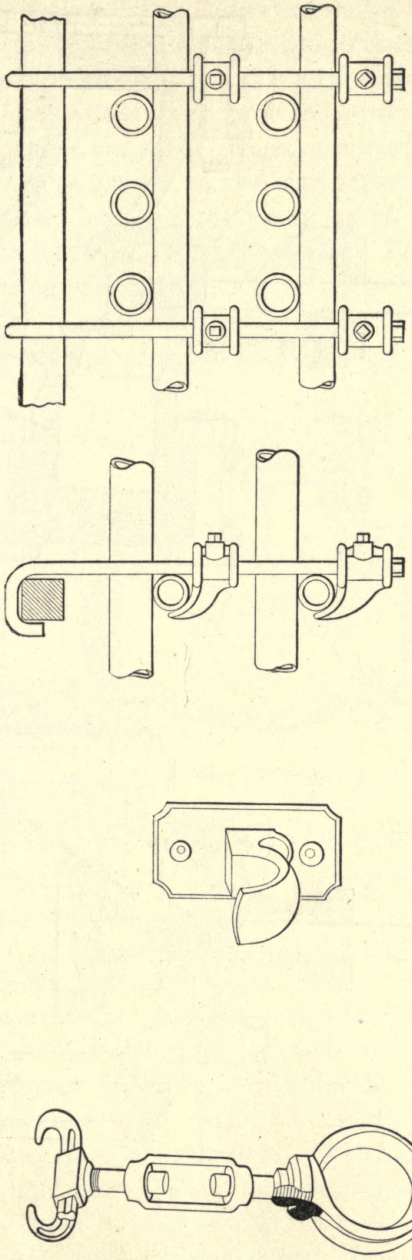


FIG. 66.—Hangers for Pipes.

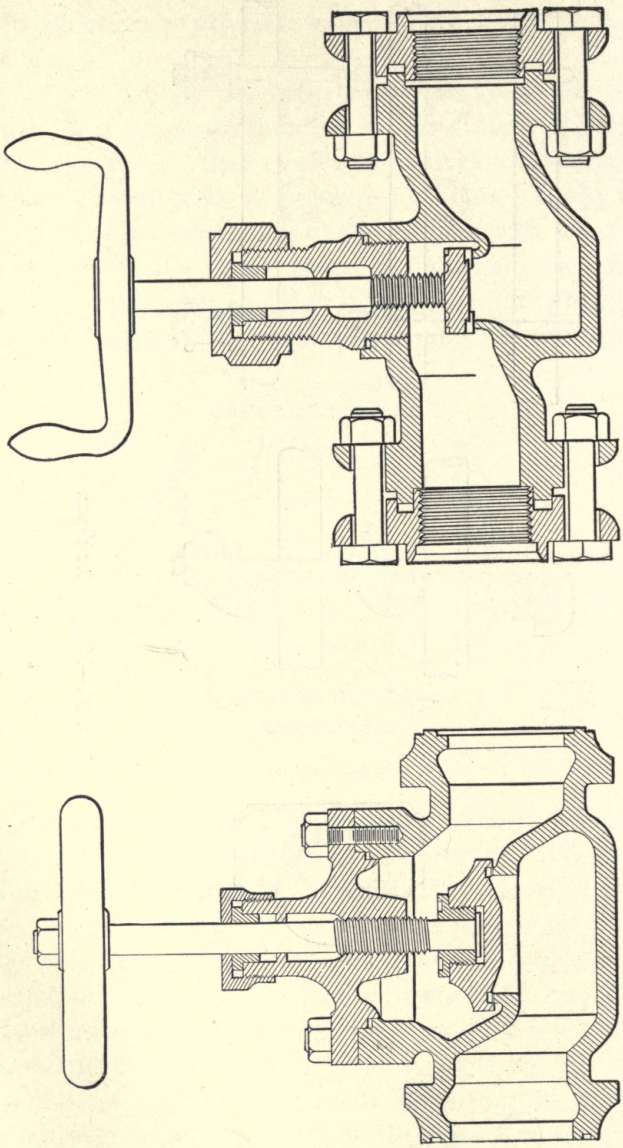


FIG. 67.—Stop Valve for Ammonia.

using a lead gasket. Two valves are shown in Fig. 67 and each of them is provided with flange connections. For angle connections, valves are built in this form, using an angle body. Where liquid ammonia or vapor is to be prevented from returning, check valves are used. These are made as shown in Fig. 68, of a lift type known as the cup pattern, due to the guide cylinder on the back, or they may be of the swing pattern. The massive construction is shown here. For expansion valves a small opening which may easily be adjusted for small changes is used. This means a needle valve and hence the forms shown in Fig. 69 are employed. In each of these the needle valve is

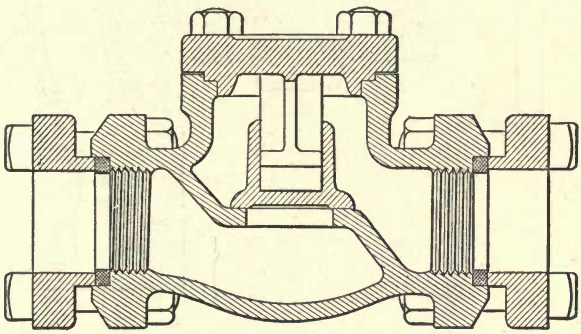


FIG. 68.—Check Valve.

raised by a fine-pitch thread so as to give close regulation on the amount of liquid discharged. Safety stop valves are built for the discharge valves of compressors. These valves are provided with a spring-closed by-pass valve which only opens when the pressure reaches a high value.

Condensers. The ammonia condensers are of various forms, depending on the plant, its location and size. An open-air surface condenser should be used when the cooling water carries scale-forming salts which would be deposited at 100° F. This condenser may be of several forms. If welded into a continuous coil there would be a difficulty in renewing a part of it. Fig. 70 shows the welded form of Kroeschell & Co. for CO_2. Welded coils are rarely used as condensers. This form is very often

used as an expansion coil. In Fig. 71, a condenser fitted with flange joints between the return bends and pipes is shown, while in Fig. 72 screwed joints are used. In Fig. 71 the

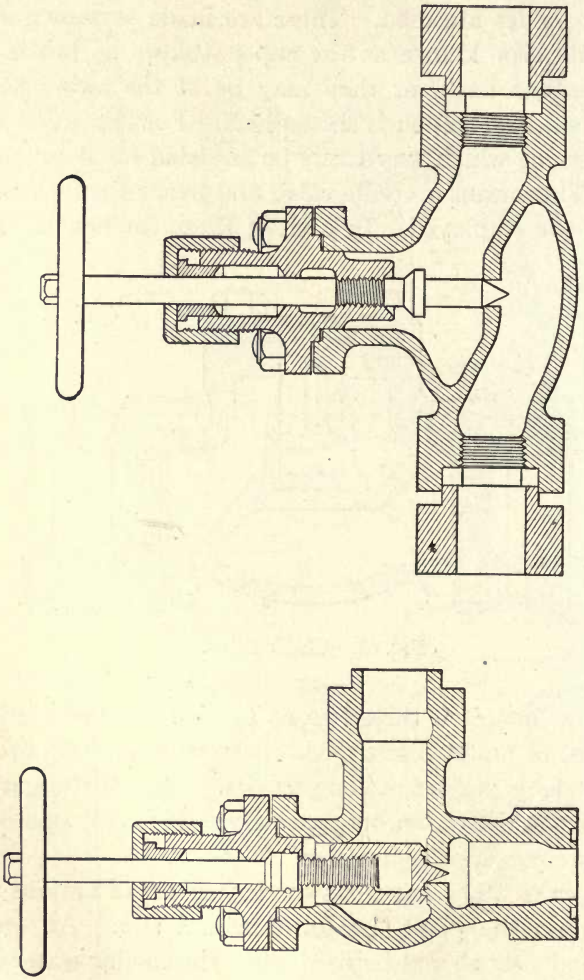


FIG. 69.—Expansion or Throttle Valves.

hot ammonia vapor passes through two lower pipes and then is taken to the top pipe in contact with the coolest water. This water is distributed from the perforated or split pipe at the top of the rack of tubes. In this arrangement there

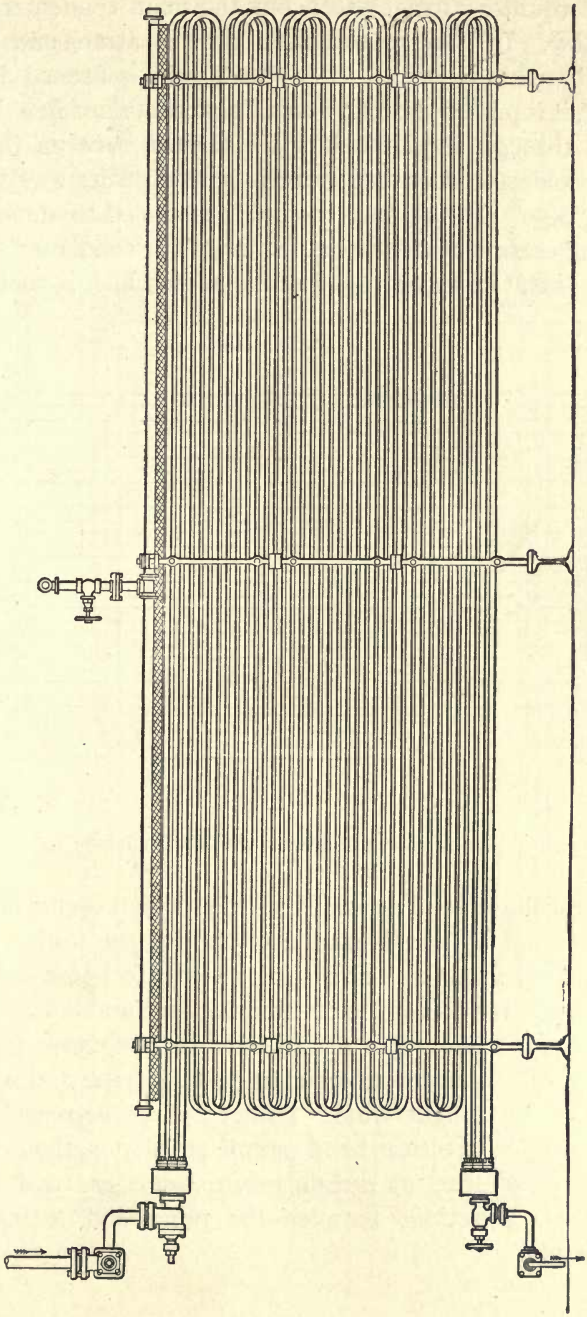


FIG. 70.—Kroeschell Welded CO_2 Condenser.

is a slight counter-current effect, but the main condenser is of parallel flow. In Fig. 72 the same general arrangement has been used with a change. After the liquid is formed in the condenser, it is passed through a small pipe contained in a larger pipe, and through the annular space formed between the two pipes the coldest condensing water is passed on its way to the sprinkling pipe. In this way the liquid is cooled to almost the lowest temperature of the cooling water. The condenser shown in Fig. 73 is one in which vapor enters at A, which is cooled by

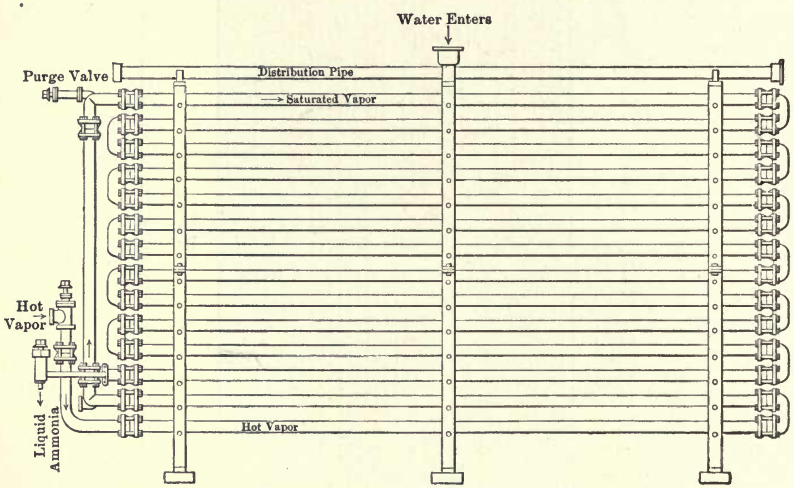


FIG. 71.—Ammonia Condenser with Flange Joints.

the warmest liquid, and any condensation which occurs is taken off at B, C and D, and is passed to the storage tank. Other liquid is taken off at E. The cooling water enters at F and flows over the slot in the top of the distributing pipe and falls over the pipes. At times plates are placed between successive pipes so that water will follow from pipe to plate to pipe and will not be blown off by a light wind. Flanged joints between elbows forming together a return bend permit an easy method of construction. The drips at certain return bends are cast in the bend. The connections between the pipes and fittings are screwed joints.

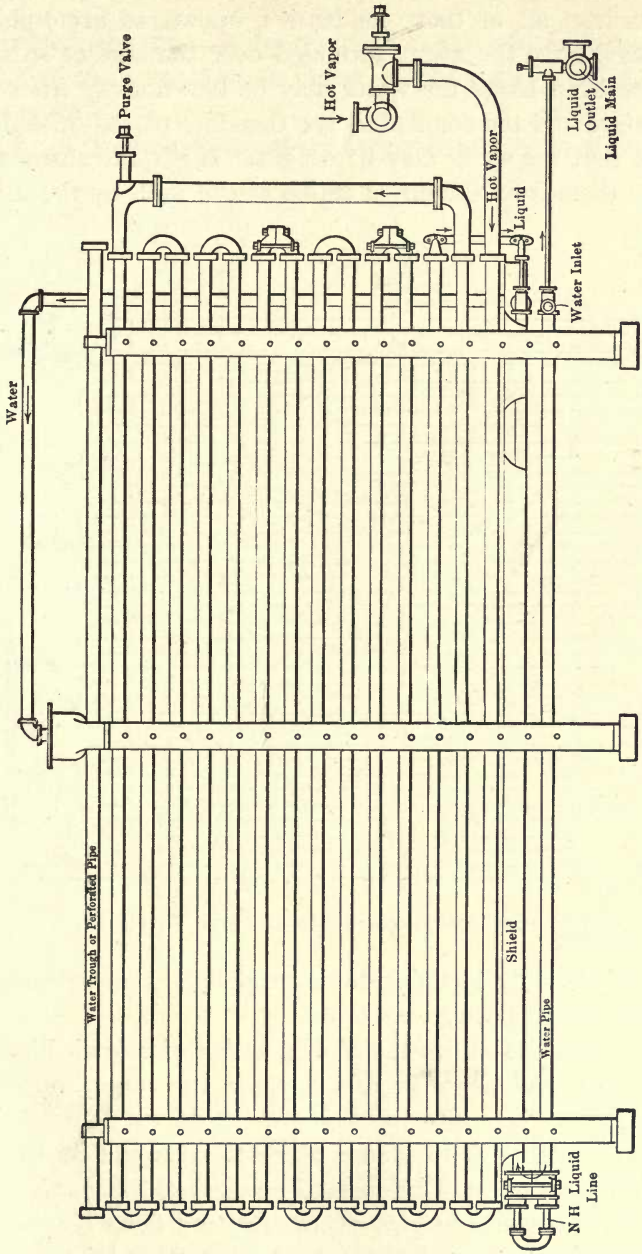


FIG. 72.—Frick Ammonia Condenser with Screwed Joints.

Of course, all of these condensers, known as atmospheric condensers, have the water sprinkled over the surface so that when the wind blows the water may be blown away from the pipe surface and the condensers are therefore placed in shallow tanks so that the water may be caught. It is quite customary to shield them from the direct action of the wind by the use of

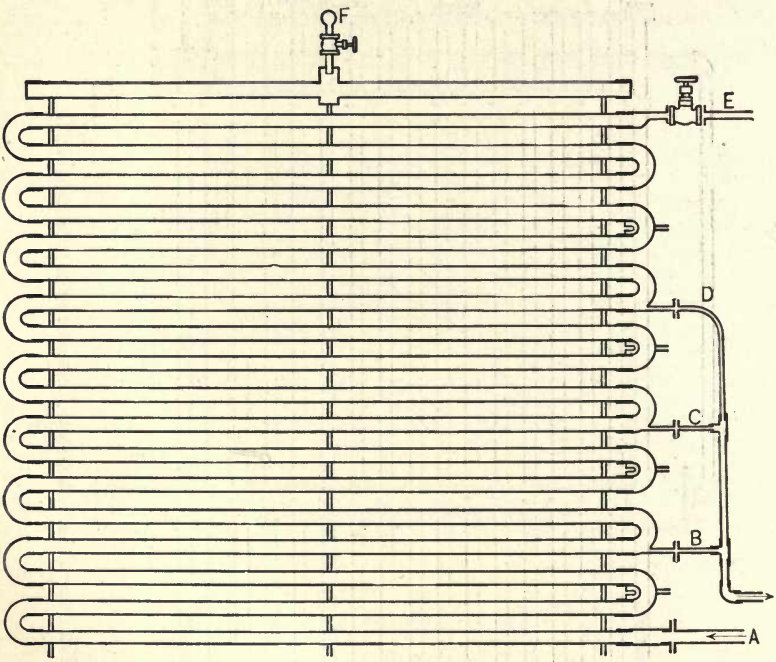


FIG. 73.—Sectional Diagram of De La Vergne Condenser.

slatted blinds. By inclining these properly the water striking them will be sent back to the tank.

The condensers utilize the cooling action of cool air blowing against them and hence in cold weather the supply of water is decreased.

In small plants or in places where there would be trouble from the falling water, a submerged condenser, Fig. 74, is used. In this the vapor is admitted at the top and flows downward. Cooling water enters at the bottom of the tank and flows to the

sewer from the top. A drain placed at the bottom will remove all water when necessary to break off the scale. This is of the counter-flow type.

One of the best forms of condenser in which the use of free-

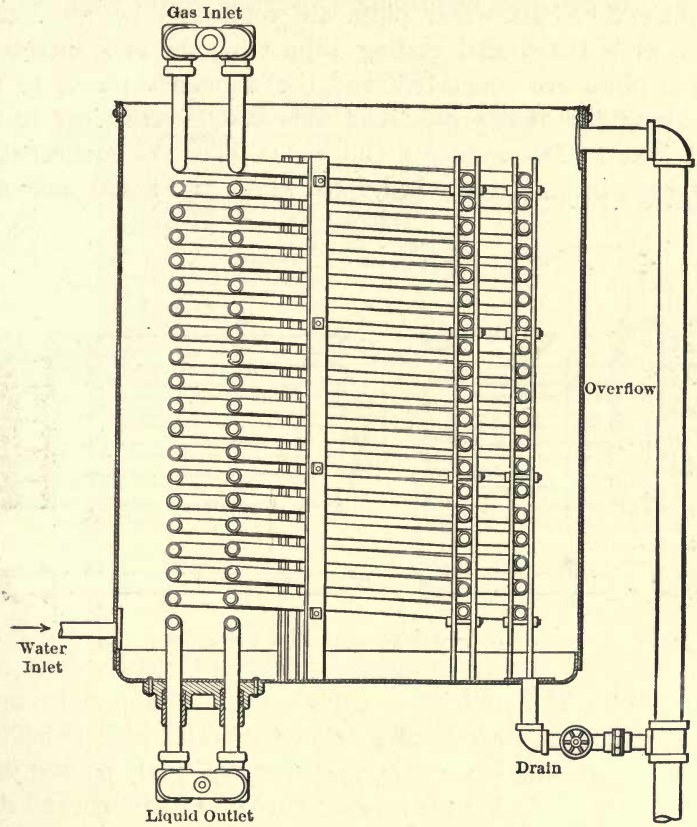


FIG. 74.—Frick Submerged Ammonia Condenser.

falling water is inadvisable and also where there is little danger of scale forming, is the double-pipe condenser, Fig. 75. In this one pipe is placed inside of another one, heavy fittings being used to connect the ends so that cooling water may be passed through the inner pipe while ammonia is condensed in the annular space between the two pipes, thus using the air as a

heat-absorbing medium as well as the water. The ammonia vapor enters at the top at *A*. The inner pipe passes through this special casting having a lead packing-ring stuffing-box. From the end *B* the warmest cooling water leaves. The ammonia passes through the annular space between the two pipes. At the end *C*, the water pipes are connected by the return bend, while the special casting supporting the two successive sets of pipes are connected and the ammonia passes to the next level and at the other end this run is connected to the next lower. The ammonia and water pipes are connected in this way until *D* is reached, from which the liquid ammonia

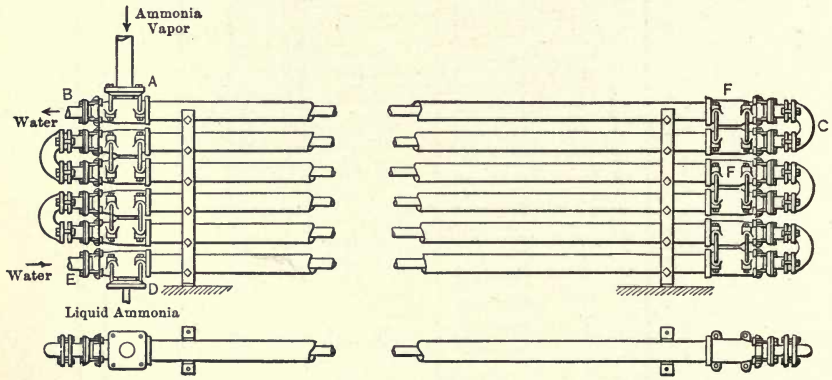


FIG. 75.—Frick Double Pipe Condenser.

passes out. The cold water enters at *E*. The special return-bend castings are arranged so that the water pipe is held in place by a stuffing-box and by means of a properly packed joint a projection of one box fits into a groove of a lower one and thus connects the ammonia channel of two successive lines. The bends are held together by bolts. By using the flanges on the outside pipe and the special casting *F*, any pipe can be removed with little work.

In this double-pipe condenser the velocity of water may be increased to a high value, thus increasing the value of the coefficient of heat transfer *K*. This is one of the important features of the double-pipe condenser. Its main use has been,

however, to remove the free water from the installation which results in dampness. This is also advisable when the water is to be used for other purposes, because the water, which is under pressure, may be delivered to any point after warning.

Fig. 76 shows the method of forming double-pipe condensers, as recently suggested by the Philadelphia Pipe-Bending Co. In

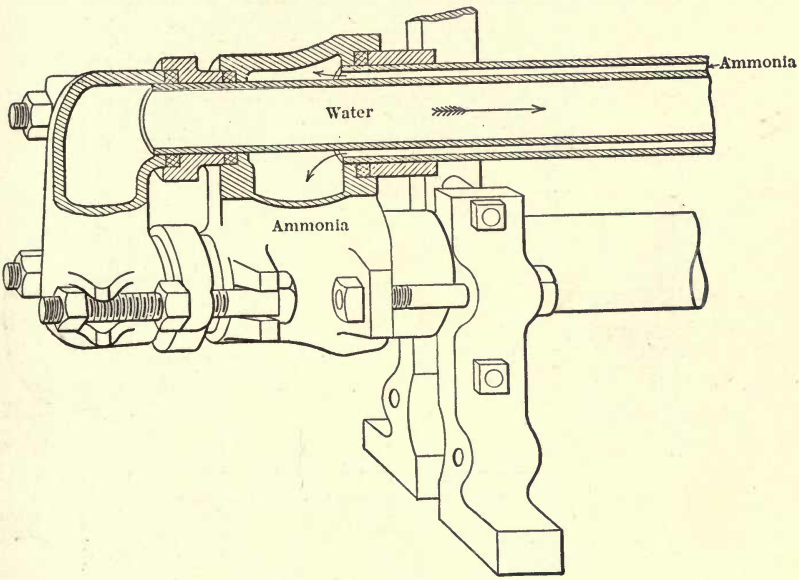


FIG. 76.—Philadelphia Pipe Bending Co. Double Pipe Condenser.

these condensers a tight joint is made by packing being forced against the pipe by pressure.

The latest improvement in condensers is to have a liquid coating on the ammonia side of the pipe, as it has been the experience of those familiar with this apparatus that if the pipes are covered with liquid ammonia they will transmit more heat per degree difference per hour per square foot than they will if not wetted. The York Manufacturing Company obtains this wet condition by injecting a certain amount of condensed liquid from the condenser back with the compressed vapor on its way to the condenser, using an injector nozzle to take up the liquid

and introduce it into the condenser with the vapor. Block accomplishes the same thing by casting a ridge in the return bends of his condenser, forming a dam which retains a certain amount of liquid in each pipe. In this way increased duty from a given amount of surface is made possible by the presence

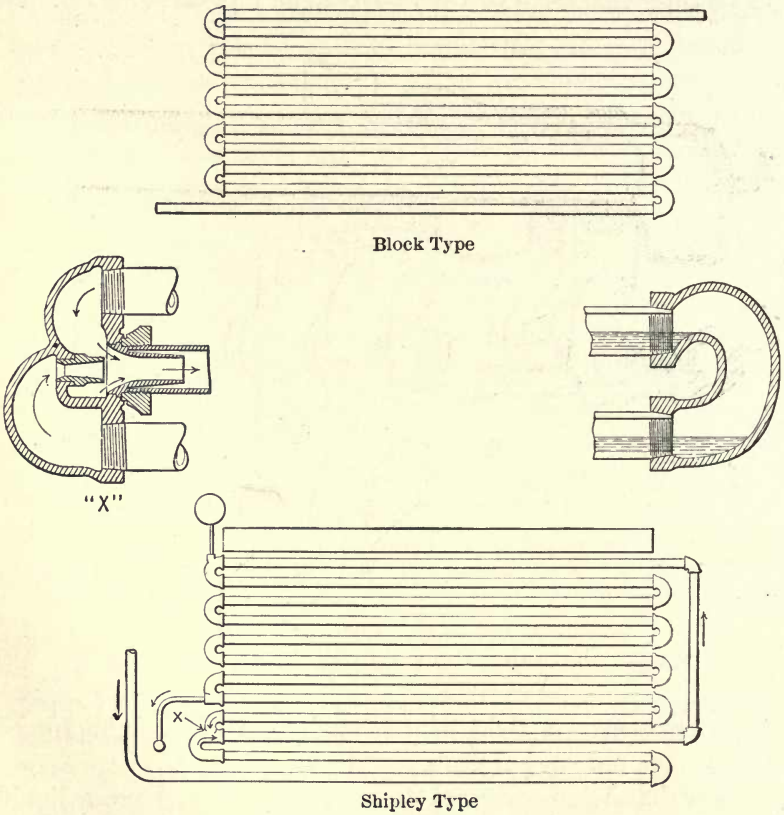


FIG. 77.—Block and Shipley Condensers.

of liquid inside. This liquid is carried along by the vapor flow.

The condenser pipes are supported by vertical pipe supports shown in Fig. 78. The pipes are either held between the supports or on brackets projecting from the side. Bolts are used as the supporting element in the right-hand type, while in the

middle form the pipes are held by the castings when they are held together by bolts.

Separators. The separators used in the ammonia systems for scale or oil separation should be of the same form as those used in engine work, but with heavy walls and flanges and a strong gasket packing. Fig. 79 illustrates the Triumph Ice Machine Co. oil separator. The incoming vapor and oil are

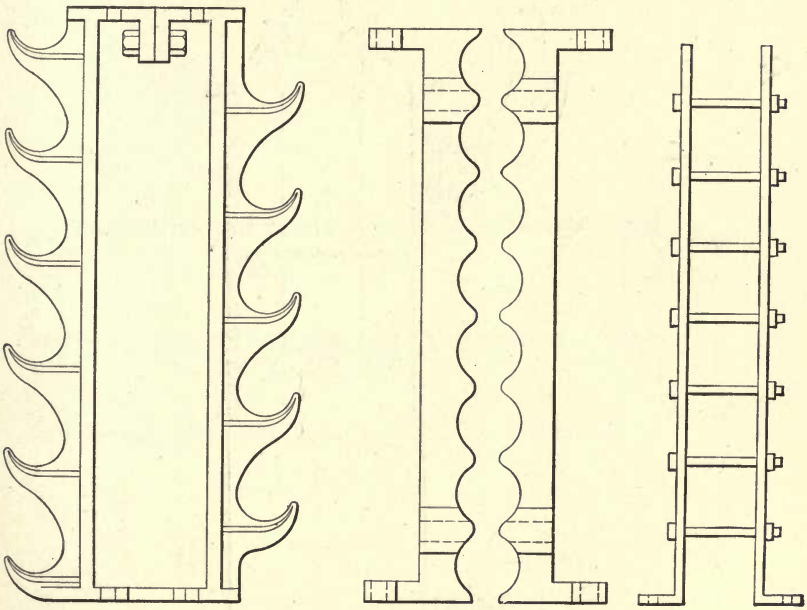


FIG. 78.—Condenser Pipe Supports.

discharged by means of a cone against the side walls of the separator and the oil will cling to the wall while the vapor rises slowly through the large cross-section of the cylinder and passes a strainer *A* and leaves at the outlet *B*. The oil and water may be drawn off at the drain valve.

Liquid Receiver. Liquid receivers are usually made of pieces of extra heavy wrought-iron pipe with flanged heads welded in. They are strong and durable. These are usually tested to 500 lbs. air pressure. The outlet from these is con-

nected to the bottom, the inlet being at the top. In some cases the discharge from the condenser enters at the outlet pipe, and if there is more liquid coming from the condenser than that required by the expander, the liquid can collect, since the two valves are connected by an equalizing pipe.

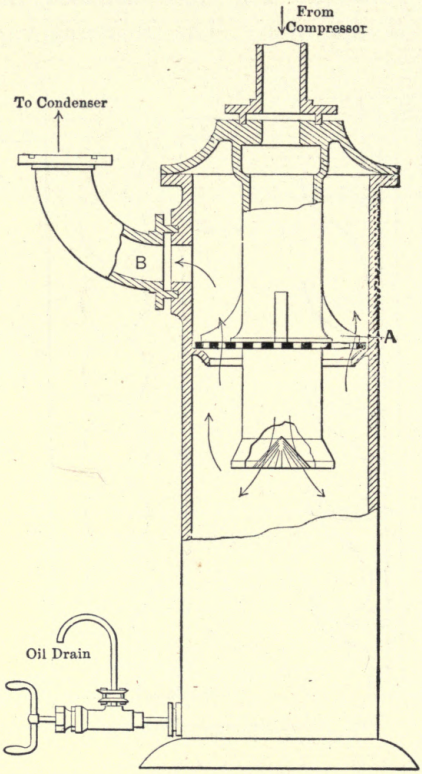


FIG. 79.—Triumph Oil Separator.

Brine Cooler. The brine cooler is an apparatus in which heat is abstracted from the brine by the evaporation of the ammonia. Usually the liquid ammonia is admitted at the bottom of a coil of pipe and the vapor resulting from the evaporation is taken off from the top. Around this coil the brine is circulated, or in some cases the brine is in the coil while the ammonia is on the outside. In Fig. 19, the brine cooler is

equipped with a brine coil, the liquid ammonia being carried about one-third of the height of the chamber of the cooler. It is always well to have the liquid ammonia in contact with the metal of the coil. It would probably yield a larger heat transfer in Fig. 19 if the liquid ammonia were sprayed over the brine coil from the top of the chamber. After the vapor is formed there can be little if any further heat removed, so that all surface in contact with vapor alone is of little value.

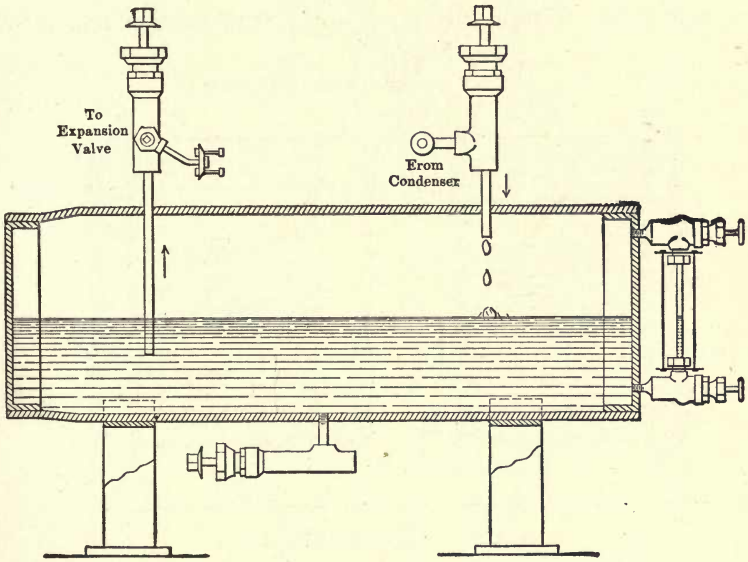


FIG. 80.—Liquid Receiver.

In the Vogt brine cooler, Fig. 81, the brine is passed through the horizontal tubes running between the two head plates in a four-pass course, while the liquid ammonia is introduced at the center of the shell. In this there is ample heat transfer and the surface is efficient.

Fig. 76 would represent a double-pipe brine cooler as well as a condenser if the liquid ammonia were placed on the inside of the coil and the brine were on the outside.

Fig. 82 illustrates a triple-pipe brine cooler. In this, liquid ammonia enters the annular space between the two inner tubes

of a set of three consecutive tubes at *A*. This space of one line is connected to the space on the next level by the special return bend *B* in the same manner as was used in the double-pipe condenser. This is then repeated at alternate ends until the outlet *C* is reached. At this point is the suction pipe leading to the compressor. The brine enters at *D* and passes through the special casting into the outer annular space and then by similar castings at *E* it enters the second row, finally reaching *F*, at which point it is connected to a return bend and enters the center of the middle tube. Finally, by pipes and return bend it reaches the

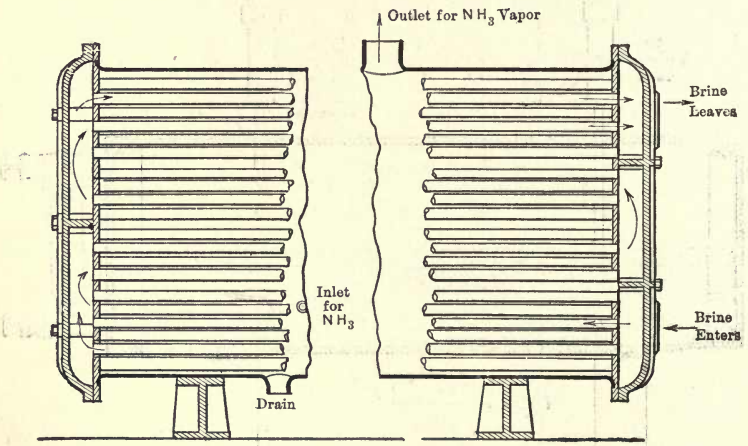


FIG. 81.—Vogt Shell Brine Cooler.

point of outlet *G*. Of course this appears to be partly counter current and partly parallel flow, but it must be remembered that the ammonia at all parts is at practically the same temperature, since there is little drop of pressure through the cooler.

The brine cooler shown in Fig. 83 is that built by the Baker Co. The ammonia lies in the inclined inner tubes and is introduced at *A*, passing down to the various pipes. The vapor is drawn through the nozzle *B*, and up to the separator *C* to the outlet *D*. Any liquid taken up is removed by the separator and sent back through *E*. Brine enters at *F* and leaves at *G*. The connections are made by return bends.

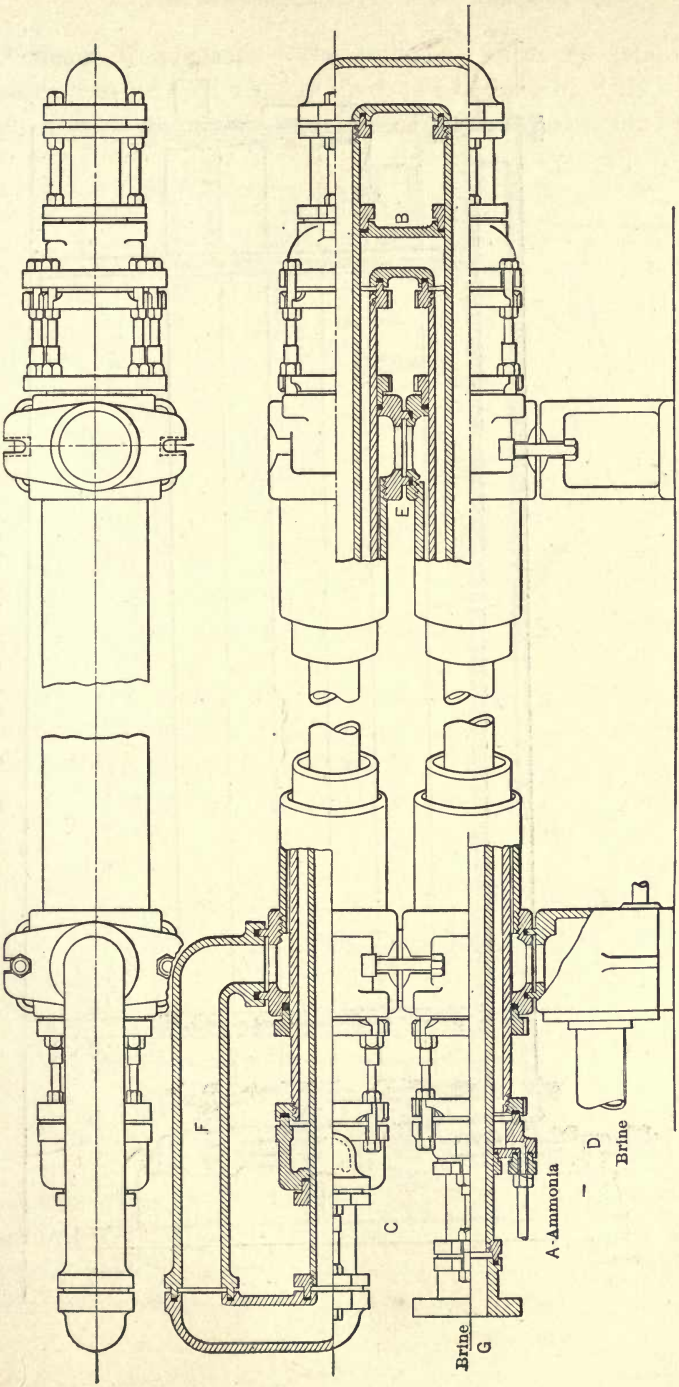


FIG. 82.—Triple Pipe Brine Cooler.

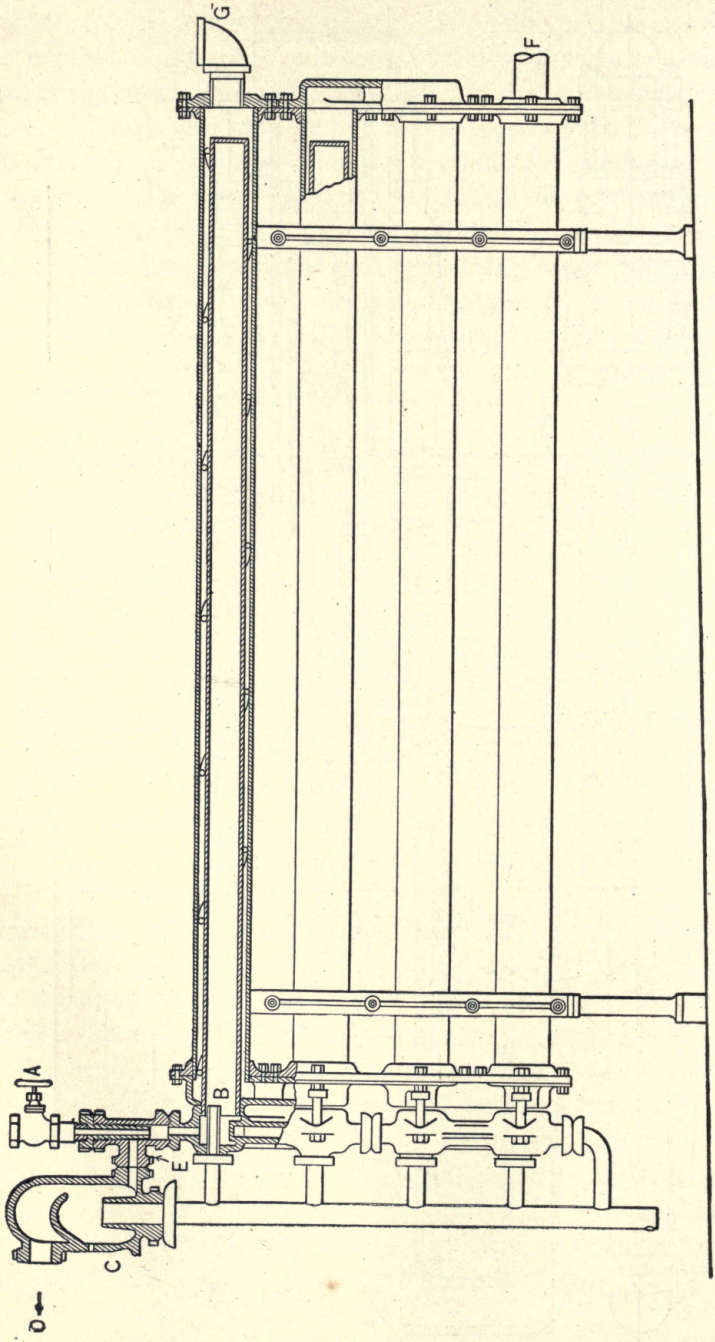


FIG. 83.—Baker Brine Cooler.

Steam Condensers. The ordinary forms of steam condensers can be used when desired, but because of scale troubles and because so much condensation is demanded for distilled

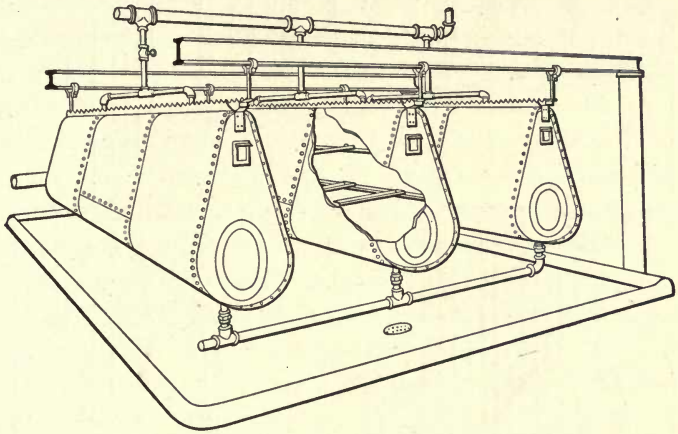


FIG. 84.—Arctic Oval Flask Steam Condenser.

water in ice plants this is done at atmospheric pressure in **flask condensers.** Fig. 84 is made of sheet iron. Water is discharged over the surface of the flask and condenses the steam within. The outside and inside surfaces of the condenser may

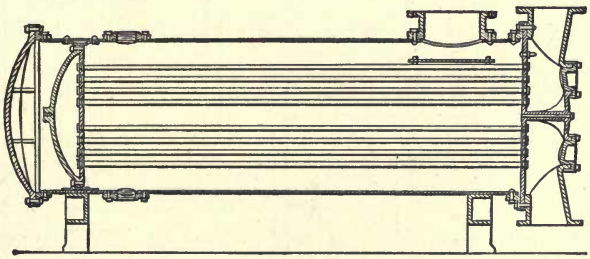


FIG. 85.—York Steam Surface Condenser.

be cleaned easily. In Fig. 85, the regular **shell type of steam condenser** is shown. The **tubes** are of brass and although usually held in place by soft packing they are expanded in the brass **tube sheets** in the figure shown. The left-hand **tube**

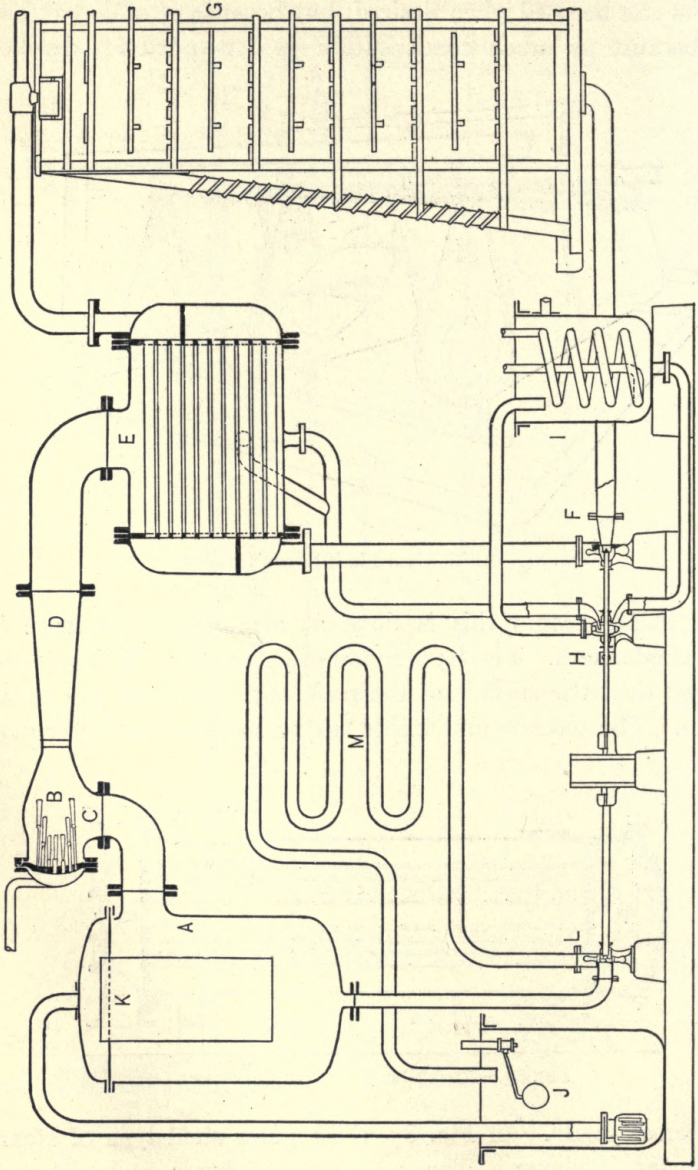


FIG. 86.—Westinghouse-LeBlanc Evaporative Refrigerating Machine. (*Power*, Jan. 11, 1916.)

plate with the water head is allowed to expand back and forth, thus caring for expansion. The steam fills the whole inner chamber of the shell and passes around the left-hand tube sheet and cap.

One of the latest developments of refrigerating machines is the **Westinghouse-LeBlanc evaporative refrigerating machine** shown in Fig. 86. This was described by Mr. J. C. Bertsch before the American Warehouseman's Association in December, 1915, and reprinted in *Power* for January 11, 1916. In this a high vacuum is maintained in the **evaporator** *A* by means of a series of **steam nozzles** *B*, from which steam at a high velocity issued, entraining with the jet any air or vapor which may be around the steam jets in the space *C*. By making the nozzles long enough and of proper shape the final pressure of the steam will be low at the end of the nozzle. The jets of steam and entrained air and vapor enter a **diffuser** *D* where the velocity is decreased and the pressure increased to such a value that the steam and vapor may be condensed by the water supply in the condenser *E*. This is supplied by the **circulating pump** *F*, the water coming from the **cooling tower** *G* which has received the warmed circulating water from *E*. The **air pump** *H* is a Westinghouse-LeBlanc centrifugal air pump and withdraws the condensation and air from *E*. The water of condensation is discharged into *I* and is cooled by a coil and flows back as sealing water for the pump.

A high vacuum existing in *B* means a high vacuum in *A*, and consequently the brine in the **tank** *J* is sucked up into this vessel and passes through the **perforated plate** *K* into an inner chamber, where it is broken up into a number of drops which, falling through the space of low pressure, are subject to evaporation of part of the water content. This of course cools the brine on account of the heat of evaporation coming from the liquid, and by the time it reaches the bottom of the evaporator it is cool enough to be circulated by the **brine pump** *L* through the **brine system** *M*, after which the warmed brine is discharged into the surge tank or receiver *J*. The brine has been concentrated in *A* by evaporation and consequently

water must be added in J to reduce the concentration by the proper amount. This is done by the float valve. The various

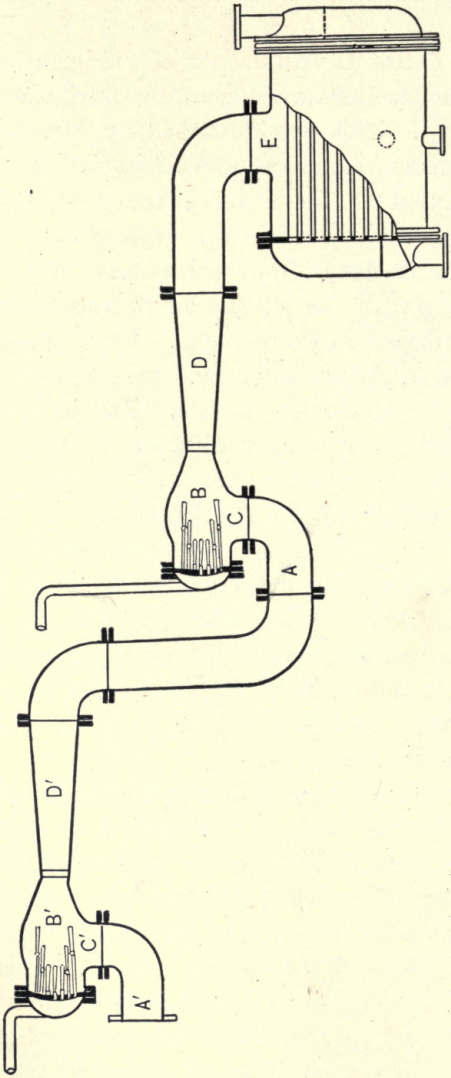


FIG. 87.—Method of Obtaining Low Pressures when Warm Water is Used.

pumps are operated on the same shaft by a steam turbine. The stuffing-boxes of the brine pump are water sealed to keep out the air.

The ejector *B*, composed of nozzles and the diffuser, are such that the vacuum carried has a temperature of boiling of 50° below the temperature of the cooling water used in the condenser. If a lower temperature is desired, say 70° to 100° below the cooling water, two of these are used in the series, as shown in Fig. 87.

Binary refrigeration is the name given to the use of a mixture of two different refrigerating media such as CO_2 and SO_2. There has been no gain shown from the use of these. The late Mr. E. Penney reports in the Transactions of the American Society of Refrigerating Engineers experiences of himself and others in this field.

In all refrigerating apparatus **the use of thermometers** is important. By their use combined with that of the pressure gauge the action of the apparatus may be known. Thus the condition of the vapor entering the compressor may be known by the temperature of the vapor at suction pressure, and by thermometers in the discharge pipe the quality of the suction vapor may be known by the amount of superheat in the discharge gas. The water and brine temperatures tell whether the surfaces are dirty. All instruments are of value in the proper operation of a plant.

Cooling Towers. Where water for condensing is scarce some method of cooling is necessary. Cooling towers are used for this purpose. In these, water is allowed to flow over screens of galvanized wire, glazed tiles, wooden slats or some other form of baffle to break the water up into small particles, and while in this condition it is brought in contact with air, which will be heated and absorb some moisture. The heating of the air cools the water and the evaporation of the water taken up by the air removes more heat. This is the principle of the cooling tower: the heating of air and the evaporation of part of the water removes sufficient heat to cool the main body of the water so that it may be used again. Of course the only place from which the heat of vaporization and heat for the air can come is the water.

Fig. 88 illustrates one form of **cooling tower.** In this hot

water is pumped through the pipe *C* to the boxes *D D* arranged at the top of the tower on the sides. This water then flows into a series of pipes *E*, which are slotted on top, from which it flows over the **mats** *B*, made of wire screens. The air

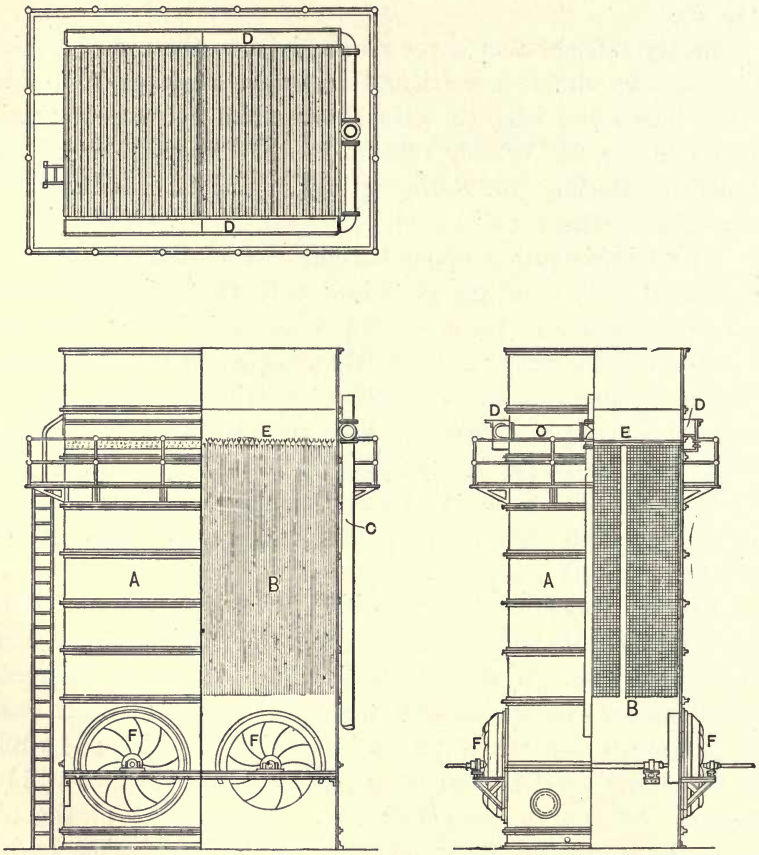


FIG. 88.—Cooling Tower.

blown in by four fans *F* meets the water falling in small drops over the screens. Here it is warmed and as its moisture capacity increases there is some evaporation. The tower proper is made of sheets of steel stiffened by angle irons.

If a sheet metal top is placed above the top in the form of a

chimney, the fans may be omitted, as the chimney effect is sufficient to cause the proper circulation of air. In some of these cooling towers glazed tiles on end form the surface over which the water is discharged. Such a tower will require about 0.2 sq.ft. of ground area per gallon of water cooled per minute.

In the **Hart Cooling Tower** shown in Fig. 89 there is no fan. The tower consists of a series of **cooling decks** *C*, *D*, *E*, *F*, spaced from 3 to 7 ft. apart, depending on the amount of cooling and the quantity of water to be cooled. The cooling decks are made up of trays placed in a staggered position on the upper and lower flanges of the I-beam supporting the deck. The water from the supply pipe *A* is delivered through a set of **spray nozzles** *B* above the first set of trays and there falls over the successive trays, a total drop of from 20 to 50 ft. The splashing of the water as it strikes the tray causes it to fall in drops. To prevent the spray from being lost the projecting shields *G* are

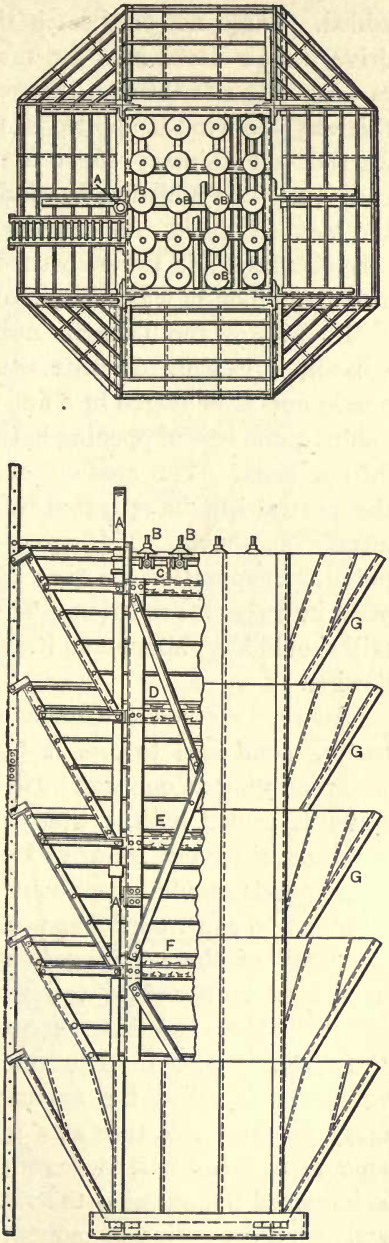


FIG. 89.—Hart Cooling Tower.

added. These not only catch the water blown away but they drive the air down into the tower when the wind is blowing, causing it to rise through the center or on the other side. In this way the wind is applied to operate the tower and in calm weather the chimney effect from the heated air causes a circulation. The moisture in windy weather is usually caught on the leeward shield and delivered to the next deck. This and other atmospheric towers occupy from 1 to $1\frac{1}{2}$ sq.ft. of ground area per gallon of water per minute.

In Fig. 90 the **Thomas nozzle,** used to spray water over a basin, is shown. In this the hot water is pumped to the nozzle and is delivered in a fine spray. The discharge orifice is made by the helical opening between the edges of a strip formed into a helix. The amount of opening may be regulated by the central spindle operated by a rod which controls all of the nozzles as shown. This spray of water warms the air and permits evaporation, so that the water falls to the **tank** or **pond** in a cooler condition. These ponds into which the spray falls should have about 2 sq.ft. of area for each gallon per minute flowing.

Another method of cooling water is to discharge it into a **cooling pond** and to cool it by surface evaporation, the hot water entering at one end. If the pond is sufficiently large the water is cooled by the time it reaches the other end of the pond or reservoir from which the cooling water is taken. These ponds should have about 70 sq.ft. per gallon of water per minute or 9 sq.ft. per horse-power hour per day.

In all of these arrangements the **cooling has been done by the heat utilized to warm air and evaporate water.** The heat to do these two things has to come from the water and hence the water is cooled. The amount of moisture which the air will carry is called the amount to saturate it. The air will carry no more moisture at a given temperature, but since the amount to saturate it varies with the temperature the capacity is increased by warming the air. Thus if at 75° 1 cu.ft. of air will carry 9.4 grains of moisture, the quantity is increased to 14.9 grains if the temperature is increased to 90°. If air at 75°

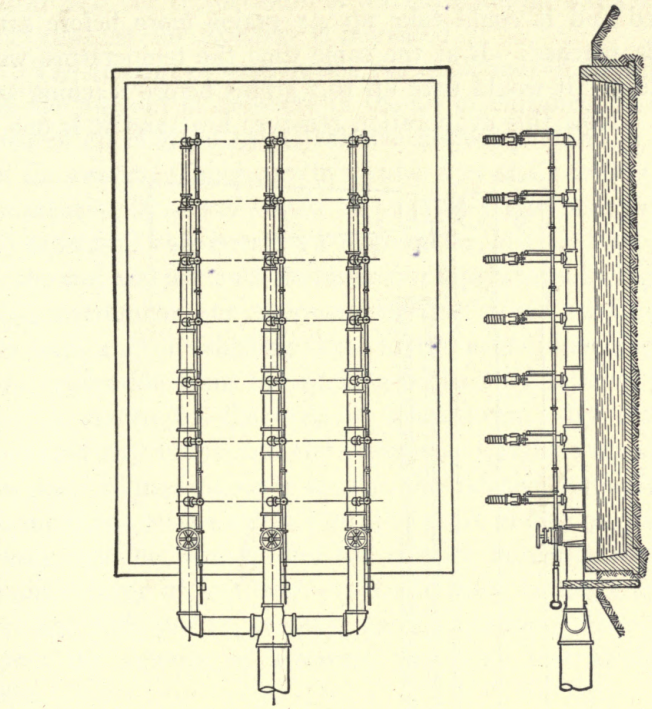
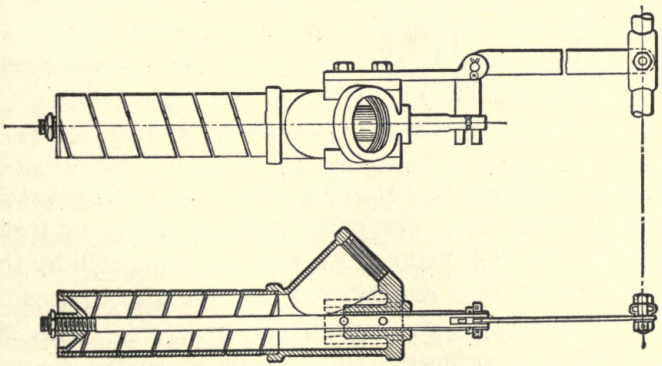


FIG. 90.—Thomas Spray Nozzle.



contains only 4.7 grains per cubic foot it is said to be one-half saturated and it could take up 4.7 grains more before saturation is reached. If at the same time the temperature were raised to 90° it would take up 10.2 grains before reaching saturation. Now this evaporation removes heat and it is one of

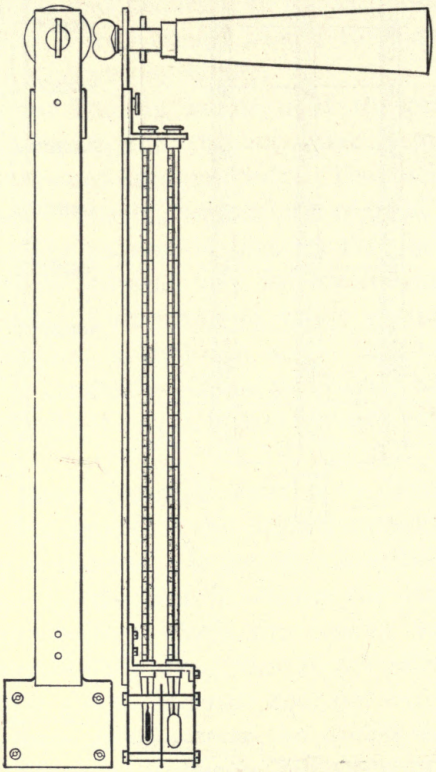


FIG. 91.—Wet and Dry Bulb Hygrometer.

the important methods of removing heat in a cooling tower. That cooling towers may evaporate water on rainy or freezing days when the air is saturated is seen to be possible by the figures above, when it is remembered that the air is raised in temperature and with it the capacity for moisture. Thus air entering at 75° and saturated will require an evaporation of 5.5 grains per cubic foot of air to saturate it at 90° if this is

the temperature of leaving. In most cases the air will leave at or near the temperature of the hot entering water and hence the capacity for moisture is increased.

The amount of evaporation per cubic foot of air will depend on the amount of moisture in the air at entrance. The condition of air is given by its **relative humidity.** This is the ratio of the amount of moisture in a cubic foot of air to that required to saturate it, as was stated on page 50. The instrument used to determine this is called a **hygrometer,** and the simplest form is the **wet and dry bulb thermometer type** shown in Fig. 91. In this instrument one thermometer has a wet wicking around its bulb and on whirling these in the atmosphere the amount of evaporation from the wicking is fixed by relative humidity and is shown by the drop in temperature. By reading the wet and dry bulbs and the barometer Carrier's equation (38) on page 51 may be used to find the relative humidity. This formula has been used to form the chart of Fig. 92, so that the figure may be used to find the relative humidity for different temperatures of wet and dry bulb, since barometric changes are not sufficient to produce much variation. The chart also gives the amount of moisture per cubic foot at any condition.

The **air required by a tower** for a given amount of water is found by equating the energy in the substances entering and leaving. It is found that the water may be cooled to a temperature much below the atmosphere in dry weather, the final temperature being about 5° above the wet bulb temperature. In times of high relative humidity of 70 or 75% the water will leave at about the atmospheric temperature, but when the relative humidity is 40% it may be from 10 to 15° below the atmosphere, showing that the evaporation of water has produced the principal cooling. The leaving temperature may be taken at about 5° above the wet bulb, although this may be 1°, 2° or even 0° above the air for high humidity.

To compute the amount of water cared for by 1 cu.ft. of air entering the following method is used:

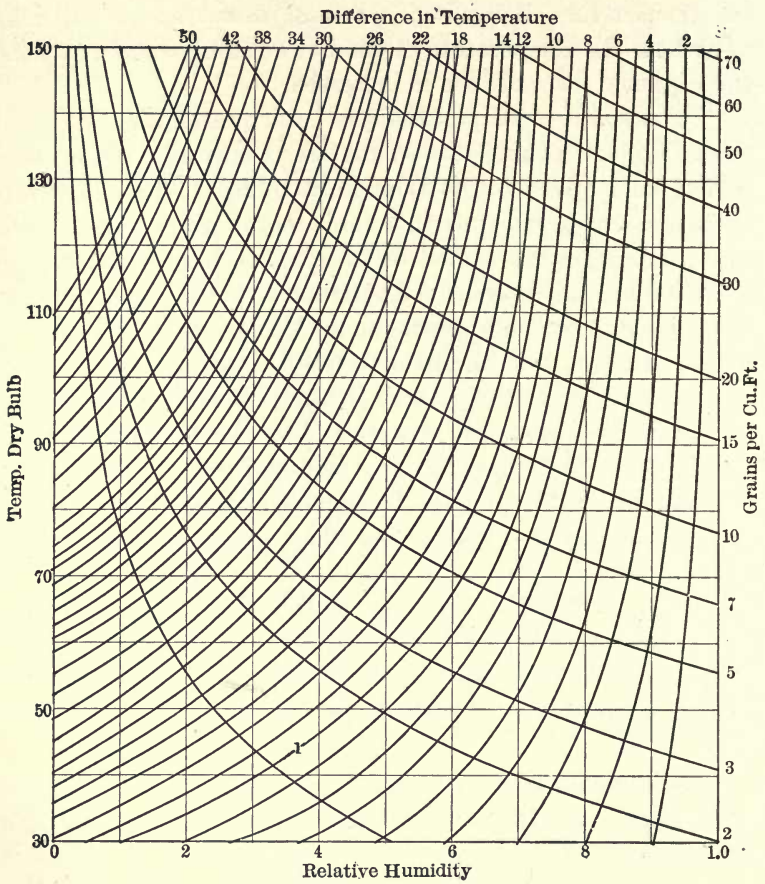


FIG. 92.—Relative Humidity and Moisture according to Carrier's Formula.

Let t_{ad} = temperature of dry bulb in air entering;
t_{aw} = temperature of wet bulb in air entering;
t'_{ad} = temperature of dry bulb in air leaving;
t'_{aw} = temperature of wet bulb in air leaving;
t = temperature of water at exit;
t' = temperature of water at entrance;
ρ = relative humidity at entrance of air;
ρ' = relative humidity at exit of air;
Bar = barometric pressure in pounds per square inch;
p = steam pressure at temperature t_{ad};

p' = steam pressure at temperature t'_{ad};
m = weight of 1 cu.ft. saturated steam at t_{ad};
m' = weight of 1 cu.ft. saturated steam at t'_{ad};
M = weight of water entering per cubic foot of air at entrance;

Assume:

$$t = t_{aw} + 5°;$$

$$t'_{aw} = t'_{ad} = t'$$

or

$$\rho' = 1.$$

Then

Moisture per cubic foot of air at entrance $= m\rho$;

Volume of air at exit per cubic foot at entrance

$$= \frac{144(\text{Bar} - \rho p)}{53{\cdot}35 \times (t_{ad} + 460)} \frac{53{\cdot}35(t'_{ad} + 460)}{144(\text{Bar} - p')} = V'. \quad . \quad . \quad (9)$$

Moisture in air at exit per cubic foot at entrance

$$= \frac{\text{Bar} - \rho p}{\text{Bar} - p'} \times \frac{t'_{ad} + 460}{t_{ad} + 460} \times m' = m''. \quad . \quad . \quad . \quad . \quad . \quad (10)$$

$$\text{Moisture absorbed} = m'' - m\rho = m'''. \quad . \quad . \quad . \quad . \quad (11)$$

Energy entering tower:

With 1 cu.ft. air

$$= \frac{144\,(\text{Bar} - \rho p)}{0{\cdot}4} + 144\,(\text{Bar} - \rho p) = \frac{1{\cdot}4}{0{\cdot}4} 144\,(\text{Bar} - \rho p). \quad (12)$$

$$\text{With } m \text{ lbs. moisture} = m\rho i J. \quad . \quad . \quad . \quad . \quad (13)$$

$$\text{With } M \text{ lbs. of water} = Mq'_t J. \quad . \quad . \quad . \quad . \quad (14)$$

Energy leaving tower:

With air of V' cu.ft.

$$= \frac{1{\cdot}4}{0{\cdot}4} 144\,(\text{Bar} - p') \frac{\text{Bar} - \rho p}{\text{Bar} - p'} \frac{t'_{cd} + 460}{t_{cd} + 460}$$

$$= \frac{1{\cdot}4}{0{\cdot}4} 144\,(\text{Bar} - \rho p) \frac{t'_{cd} + 460}{t_{ad} + 460}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

With m'' lbs. of moisture $= m'''i'J$. . . . . (16)

With $(M - m''')$ lbs. of water $= (M - m''')q'J$. . . (17)

In these i' is the heat content of vapor in saturated or superheated condition and q' is the heat of liquid.

By equating these the quantity M may be found. This gives the **amount of water per cubic foot of air entering,** or the reciprocal will give the cubic feet of air per pound of water entering. In this way the air for a given amount of water may be found and the fan to introduce this air or the chimney to suck this air may be computed. The evaporation per pound entering, $\frac{m'''}{M}$, will give the amount of make-up necessary.

In designing nozzles for cooling fountains the assumption may be made that the evaporation will reduce the temperature provided the final temperature is above the wet bulb temperature. The number of nozzles may be found from the quantity of water by using the following data:

The capacity of the spray nozzles of the Thomas form is 150 gallons per minute under 4 lbs. per square inch pressure. For ordinary spray nozzles the discharge in gallons per minute is given by the table:

Size, inches.	5 lbs. pressure.	10 lbs. pressure.
1.5	15	21
2.0	40	60
2.5	70	90
3.0	120	140

A velocity of 5 ft. per second at the entrance to the nozzle will give the discharge at 5 lbs. pressure. The tank or reservoir used with these nozzles should have about $2\frac{1}{2}$ sq.ft. of surface for every gallon per minute. When the weather is warm a second spraying is necessary in successive nozzles over separate reservoirs.

The **cooling pond** may be made of such a size that 1 sq. ft. will care for $3\frac{1}{2}$ B.T.U. per hour per degree difference in tem-

perature between air and water; 70 sq.ft. per gallon per minute of water to be cooled has been suggested for the area of the pond.

Safety Devices. There are several devices which are installed in all plants using poisonous or suffocating gases to prevent loss of life and make the operation of the plant possible. One of the important ones is a **helmet** to wear over the head when it is necessary to enter a room filled with fumes to repair a break,

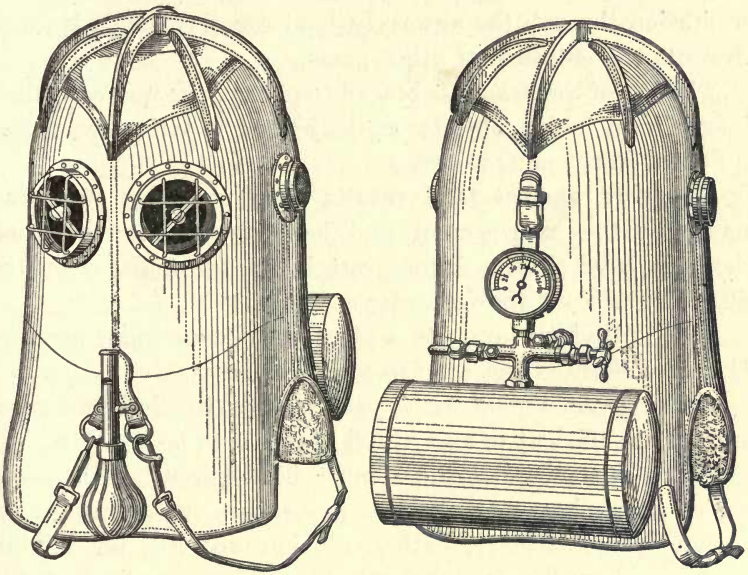


FIG. 93.—Improved Vajen Helmet.

to shut a valve, or to rescue a person. There are several of these in use. One of them is shown in Fig. 93. The Vajen helmet weighs 10 lbs. and fits over the shoulders, being strapped tight on a wool gasket. The weight is carried by the shoulders, leaving the head free to turn. The air contained in the reservoir under pressure is sufficient for one-half to one and one-half hours' use. The helmet is made of fire and water-proof materials, and by the large double-plate mica-covered openings guarded by cross bars one can easily see to work. Rotary cleaners are provided to clean these if obscured by smoke or

moisture. Telephonic ear pieces with special sounding diaphragms enable the operator to hear distinctly and a whistle on front of the helmet makes it possible for the operator to signal others.

The air reservoir is charged in two minutes by an air pump, and although this is a short time the reservoir should be kept charged. By opening the valve on the top reservoir air is discharged into the helmet in front of the nostrils of the wearer. This is above atmospheric pressure and forces the gases from respiration through the absorbent lambs-wool collar gasket and prevents the entrance of other gases.

The top of the helmet is braced to protect the wearer against danger from falling objects, as the helmet is intended for use in fires or in chemical works.

To guard against fatal results from accidents rules are made for the management and installation of refrigerating plants in large cities. Among certain rules formulated by the city of New York the following are noted:

It is unlawful to operate a plant with gases under pressure without a license from the fire commissioner.

An emergency pipe with valve outside to discharge gases into water sufficient to absorb full charge is to be installed.

All refrigerating machines must be equipped with safety valves to discharge at 300 lbs. per square inch pressure for ammonia, 1400 lbs. for carbon dioxide, 100 lbs. for sulphur dioxide and 100 lbs. for ethyl chloride. These are to discharge into emergency pipes or to the low-pressure side of the system.

There must be provisions for exit into the outside air or to a hall from which gas can be excluded for all rooms when the pressures are above the following limits:

Ethyl chloride	40	lbs. per square inch
Sulphur dioxide	60	"
Ammonia	100	"
Carbon dioxide	500	"

All fittings are to be tested to twice the maximum pressure and pipes to three times the maximum.

No open flames are allowed in any rooms having pressure pipes.

Helmets must be installed in all plants.

Pipes are to be tagged showing kind of substance within.

Storage of extra refrigerating substance will not amount to more than 10% of capacity. The cylinders cannot be kept in the boiler room but in some cool place.

When the plant has a capacity of more than 3 tons the operator must have a certificate.

The United States Interstate Commerce Commission has provided certain **rules relating to the cylinders** for the shipment of gases under pressure. Some of these are as follows:

Cylinders must comply with requirements and be made of lap-welded pipe of soft steel of best welding quality. They may be made seamless. The heads should be welded in. The carbon is limited to 0.20%, phosphorus 0.11%, and sulphur 0.05%. The cylinders must stand 1000 lbs. per square inch in a water jacket to give extension which must not be over 10% of volume. Cylinders must stand flattening out. They must be annealed. The cylinders must be stamped with name of owner. Gases which combine may not be shipped in one cylinder. Each cylinder must be tested once in five years under pressure. Test pressure is one and one-quarter times the pressure of vapor at 130° F., except for carbon dioxide cylinders, which are tested to 3000 lbs. per square inch.

CHAPTER V

HEAT TRANSFER, INSULATION AND AMOUNT OF HEAT

HEAT is transferred by **radiation, convection** and **conduction.** In the **first method** a body starts a vibration in the ether, which is transmitted by it to another body which receives this energy of vibration and changes it into heat energy of the body. This form of transmission depends on the difference of the fourth powers of the temperatures of the two bodies. Although important in some cases, radiation does not play an important part in refrigeration. In the **second method,** particles of some medium, as air, are heated by a hot body and then by the bodily transfer of these heated particles the energy received by them is carried to a cooler body, which abstracts the heat. This method of heat transfer is one used for some types of refrigeration. The **third method** is that in which heat energy is applied to the molecules of one part of a body, and then by transmitting this energy to adjacent molecules the energy is gradually conveyed through the body. It is in this manner that heat is taken from cold storage rooms by the brine or ammonia, or heat is added to the room from the atmosphere. The last two methods of heat transfer must be examined in detail.

If M lbs. of substance are **heated at constant pressure** from a temperature t_1 to t_2 when the specific heat is c_p, the heat required to do this is

$$Q = Mc_p(t_2 - t_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

or

$$Q = M\int_{t_1}^{t_2} c_p dt. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The first is correct if c_p is constant, or is the mean value of c_p, and the second is correct if c_p is a variable. The substance

usually employed is air, and although c_p for air is not constant, the variation for the temperature ranges used in refrigerating problems is negligible. Hence the first formula may be used with a value of 0.24 for c_p. If now the air is conducted to a room or cooler and is brought back to the temperature t_1, the same amount of heat must be abstracted, and so the heat Q taken from the first body has been given to a second body and the air or carrying medium has been left in its original condition. This heat has been carried by **convection.**

In **conduction** the heat transmitted depends on the material, the temperature, the cross-section of the material and the length of the path. The equation for conduction is similar to that for the flow of the electric current.

$$Q = \frac{CF(t_2 - t_1)}{l} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

F = area of cross-section in square feet;
l = length of path, or thickness, in feet;
t = temperature on either side in deg. F.;
C = constant of conduction, or B.t.u. per hour per square foot per degree for 1 ft. thickness.

The value of C has been determined by various experiments, and by its use the amount of heat conducted can be predicted.

When heat is transmitted through partitions, it is difficult to compute the amount of heat transmitted because it is hard to determine the temperatures at the edge of the plate on account of films of fluid which cling to the surface and make heat transmission difficult. Thus, if heat from the gases of a boiler is to be taken into the water in contact with the tube, and Eq. (3) is used to compute the probable heat transfer, using the C for steel of thickness l, and substituting the temperatures of the gas and boiling water, the heat would be equal to more than 250 times the amount actually transmitted. The great reduction is due to the effect of the films of gas and water which cling to the sides of the tube and cut down the heat transmitting power. A thin film of gas or water has a much greater

resistance than a thick wall of metal. The transmission through the gas or water film could be computed if the thicknesses were known, but these are quantities which vary, depending largely on the velocity of the fluids over the surface, and also upon the viscosity of those fluids. On account of this action experiment has been resorted to to determine a **constant** K, which is the amount of heat transmitted per square foot per hour per degree difference in temperature for the surfaces transmitting heat. Since in many cases the surface effects are the controlling factors, and not the main body of the partition, the thickness of the partition will not enter into the expression. With K the **equation for heat transmitted** becomes

$$Q = KF(t_2 - t_1); \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

K = B.t.u. per square foot per hour per degree;
F = area in square feet;
t = temperature in degrees F.

The value of K is now determined experimentally for the transmission of heat through metal walls in condensers, absorbers and such apparatus, and it is found that as the velocity of the liquids on either side increases the value of K increases. It is also found that for greater differences in temperature the value of K changes, becoming less usually as the temperature difference increases. For the temperature differences usually found in ammonia and steam condensers, brine coolers, feed-water heaters and such apparatus, the value of K found in experimental work with a given set of temperature conditions may properly be used with other temperature conditions since there would not be enough change in temperature difference to affect the value of K.

If K is a constant and the temperatures on one or both sides change along the length of the surface it is necessary to find the mean temperature difference in terms of the temperatures at the ends of each surface. Suppose that t_{h_1} and t_{h_2} are the temperatures on the warm side at entrance and exit and that t_{c_1} and t_{c_2} are the temperatures at the same ends on the cool

side of the surface. At any point x the temperatures on the two sides will be t_{hx} and t_{cx}. The amount of heat transmitted through the differential surface at this point measuring from first end will give a rise in temperature of dt_c to the cool substance of M_c pounds per second, and there will be a drop dt_h on the warm side where M_h lbs. per second flow. Hence

$$KdF(t_{hx}-t_{cx}) = -3600M_h c_h dt_h = \pm 3600M_c c_c dt_c. \quad . \quad (5)$$

As dF increases dt_h decreases and dt_c increases for parallel flow but decreases for counter current flow. The upper sign refers to parallel current flow.

$$K\Delta t_x dF = -3600M_h c_h dt_h = \pm 3600M_c c_c dt_c,$$

$$dF = \pm 3600\frac{M_c c_c}{K}\frac{dt_c}{\Delta t_x}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Now

$$M_h c_h[t_{h_1}-t_{h_2}] = \mp M_c c_c[t_{c_1}-t_{c_2}]$$

or

$$\frac{M_h c_h}{M_c c_c} = \mp\frac{t_{c_1}-t_{c_2}}{t_{h_1}-t_{h_2}}.$$

Also

$$M_h c_h[t_{h_1}-t_{hx}] = \mp M_c c_c[t_{c_1}-t_{cx}].$$

$$t_{hx} = t_{h_1} \pm \frac{M_c c_c}{M_h c_h}t_{c_1} \mp \frac{M_c c_c}{M_h c_h}t_{cx} \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$\Delta t_x = t_{hx}-t_{cx} = t_{h_1} \pm \frac{M_c c_c}{M_h c_h}t_{c_1} - \left[1 \pm \frac{M_c c_c}{M_h c_h}\right]t_{cx}.$$

$$d\Delta t_x = -\left[1 \pm \frac{M_c c_c}{M_h c_h}\right]dt_{cx}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

$$dt_{cx} = \frac{d\Delta t_x}{-\left[1 \pm \frac{M_c c_c}{M_h c_h}\right]}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$dF = \mp\frac{3600M_c c_c}{K\left[1 \pm \frac{M_c c_c}{M_h c_h}\right]}\frac{d\Delta t_x}{\Delta t_x}. \quad . \quad . \quad . \quad . \quad . \quad (10)$$

$$F = \mp \frac{3600 M_c c_c}{K\left[1 \pm \frac{M_c c_c}{M_h c_h}\right]} \log_e \frac{\Delta t_2}{\Delta t_1}. \quad . \quad . \quad . \quad . \quad (11)$$

$$F = \mp \frac{3600 M_c c_c}{K\left[1 - \frac{t_{h_1} - t_{h_2}}{t_{c_1} - t_{c_2}}\right]} \log_e \frac{\Delta t_2}{\Delta t_1}$$

$$= \mp \frac{3600 M_c c_c (t_{c_1} - t_{c_2})}{K[t_{h_2} - t_{c_2} - (t_{h_1} - t_{c_1})]} \log_e \frac{\Delta t_2}{\Delta t_1}. \quad . \quad . \quad (12)$$

$$F = \frac{\log_e \frac{\Delta t_2}{\Delta t_1}}{\Delta t_2 - \Delta t_1} \times \frac{\text{Heat added}}{K}. \quad . \quad . \quad . \quad . \quad (13)$$

Now

$$\text{Heat} = FK(\text{mean } \Delta t).$$

Hence

$$\text{Mean } \Delta t = \frac{\Delta t_2 - \Delta t_1}{\log_e \frac{\Delta t_2}{\Delta t_1}} \quad \text{or} \quad \frac{\Delta t_1 - \Delta t_2}{\log_e \frac{\Delta t_1}{\Delta t_2}}. \quad . \quad . \quad . \quad . \quad (14)$$

This then is the value for mean Δt to use in formula (4) for the solution of heat transfer problems.

If K has the value $K = \frac{K'}{(\Delta t)^n}$. . . . . (15)

$$dQ = \frac{K'}{(\Delta t_x)^n} \times \Delta t_x \times dF.$$

This leads to

$$\text{Mean } \Delta t = \left[\frac{n(\Delta t_1 - \Delta t_2)}{(\Delta t_1)^n - (\Delta t_2)^n}\right]^{\frac{1}{1-n}} \quad . \quad . \quad . \quad (16)$$

$$Q = K' \times (\text{mean } \Delta t)^{1-n} \times F. \quad . \quad . \quad . \quad (17)$$

The values of K given by various authorities are as follows:

$K = 3$ from gas to gas;
$= 5$ from liquid to gas;
$= 250$ from liquid to vapor;
$= 400$ from liquid to liquid.

The average values can be used in case of need, but more exact values are given below. These show the effect of velocity and also show the need of special experiments.

Values of K. For **ammonia condensers** of the **double type** form, the values of K as determined by R. L. Shipman in the

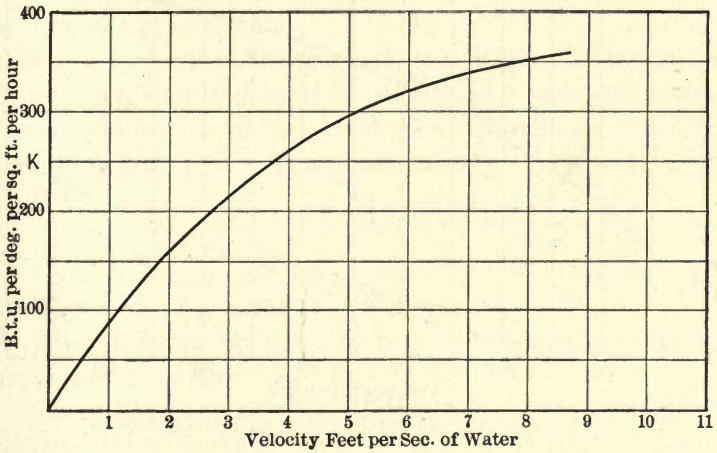


FIG. 94.—Shipman's Values of K for Double Pipe Condensers.

Transactions of the American Society of Refrigerating Engineers for 1907 are given in Fig. 94. The values of K may be taken from these curves. In Fig. 95 the values of K for **brine coils** are given at different velocities. The first of these curves has the equation

$$K = 130\sqrt{W_w} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

W_w = velocity of water in feet per second.

The equation of the second curve for **brine coils** is

$$K = 60\sqrt{W_w} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

Some allow 100 for K for double-pipe coolers and condensers.

Fred Ophuls has stated that his experiments indicate that for **double-pipe condensers**

$$K = 32.4\sqrt{W_m}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

W_m = relative mean speed of ammonia and water in feet per second. The velocity of the ammonia may be taken as one-half the velocity of the vapor at inlet to condenser.

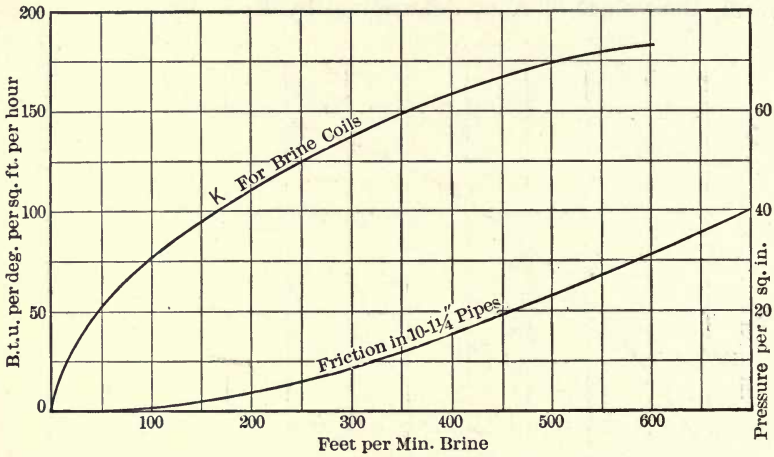


FIG. 95.—Shipman's Values of K for Brine Coils.

He also gives the following:

For **condensers of the Block type** and for **atmospheric single-pipe condensers** the experimental results were given by

$$K = 26.8\sqrt{W_v}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

W_v = velocity of vapor at entrance in feet per second.

For the cooling of the superheated vapor the constant falls off, being

$$K = 8.33\sqrt{W_{ms}}. \quad . \quad . \quad . \quad . \quad . \quad (22)$$

W_{ms} = mean speed of superheated vapor in feet per second.

Thomas Shipley gives 60 as the **value of K for open atmospheric condensers** and 300 as the value in the Shipley con-

denser, in which part of the liquid ammonia is forced through with the incoming vapor so as to wet the inside surface of the condenser. (See Fig. 77.)

Rules used in practice may be mentioned. One requires 40 sq.ft. of open air condenser surface per ton of refrigerating capacity and 25 sq.ft. for submerged condensers. These rules are to be used as checks. In many cases 18 sq.ft. are used per ton while Block condensers have been operated successfully at 8 sq.ft. per ton.

For **brine coolers and brine tanks** the value of K would be 50, although experiment must be made to find the effect of velocity more accurately.

For brine coolers Fred Ophuls and V. R. H. Greene found that although the values of K from their experiments were not correctly given by any equation, their experiments when there is no superheat were best represented by

$$K = 20.6\sqrt{0.521 W_a + W_b}. \quad . \quad . \quad . \quad . \quad (22a)$$

W_a = velocity of ammonia vapor at outlet in feet per second;
W_b = velocity of brine in feet per second.

When this ammonia leaves in a superheated condition the coefficient is changed to 15.2.

Levey gives as a rule the allowance of 55 sq.ft. of expansion coil per ton of refrigeration with at least 60 cu.ft. of capacity of brine in tank per ton. This is only a check on the computation for K. The York Mfg. Co. uses 108 sq.ft. of direct expansion coil per ton of capacity. This is 250 lineal feet of $1\frac{1}{4}$-in. pipe to ton. With ordinary piping, without flooding the coils with ammonia, 350 ft. of pipe may be used. Some have found that the formation of ice around the expansion coil in a brine tank increases the rate of heat transmission; 1 in. of ice increased the transmission to 1000 B.t.u. per foot per hour of $1\frac{1}{4}$-in. pipe and 2 ins. trebles this.

For liquid fore-coolers the constant is given by

$$K = 1.227W. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

W = velocity of the liquid in feet per second.

The value of K for coils in rooms from brine or ammonia to air, should be taken as about five, although Siebel states that ten could be used. Since ice and snow are deposited on pipes in refrigerated rooms, these constants cannot be used and the allowances employed in practice are given. These are as follows:

For direct-expansion pipes (Siebel)	1 ft. of	2-in.	pipe for	10	cu.ft.	of space at	10° F.
	1 "	2 "	"	40	"	"	32
	1 "	2 "	"	60	"	"	50
	1 "	1¼ "	"	16	"	"	35
For brine pipes (Siebel)	1 "	2 "	"	6	"	"	10
	1 "	2 "	"	26	"	"	32
	1 "	2 "	"	60	"	"	50
For brine pipes (Levey) small rooms	1 "	2 "	"	1	"	"	0
	1 "	2 "	"	4	"	"	10
	1 "	2 "	"	8	"	"	32
Room 1000 to 10,000 cu.ft.	1 "	2 "	"	2	"	"	0
	1 "	2 "	"	6	"	"	10
	1 "	2 "	"	14	"	"	32
Rooms over 10,000 cu.ft.	1 "	2 "	"	3	"	"	0
	1 "	2 "	"	9	"	"	10
	1 "	2 "	"	16	"	"	32

About 50 sq.ft. of pipe will care for 1 ton of refrigeration and 100 sq.ft. of pipe for 1 ton of ice manufactured.

Insulation

When the thickness of insulation is great or when the resistance of the internal portions becomes appreciable when compared with the resistance at the surface, the Eq. (4) is used to find the heat transmitted by the partition, and K is computed for the various elements of the partition. This is the problem of **heat transfer through walls.**

The **heat loss from rooms** is made up of several parts. There are **radiation** and **conduction** from walls, windows and doors and **convection** losses due to warming of the leakage air, or the air for ventilation. There is a gain of heat derived from persons or apparatus used in the room, or from sources of light of various kinds. The heat loss through walls partakes of the nature of radiation and conduction. The principal loss is made up of transmission, which is found to depend on the

difference of temperature and therefore it is similar to conduction rather than radiation, which depends on a higher power of the temperatures. The general form in which this heat loss is given is

$$H = KF(t_i - t_0), \quad \ldots\ldots (24)$$

where F = area in square feet;

K = heat transmitted per square foot per hour per degree difference of temperature in B.t.u.;

t_i = room temperature in degrees F.;

t_0 = outside temperature in degrees F.;

H = B.t.u. transmitted per hour.

The value of K depends upon several factors: the surface, thickness and kind of material, air spaces and condition of air at surface. It also depends on temperature difference, but since the temperature differences are not large, this effect may be neglected. The following German method from H. Rietschel's Leitfaden zum Berechnen und Entwerfen von Luftungs- und Heizungs-Anlagen is usual for future reference for cases which have not been calculated in the text.

The **rate of transmission** of heat through any substance depends upon the thickness and on the difference of temperature. If for instance the wall shown in Fig. 96 is made up of several thicknesses, and the temperatures are those marked, the equations for the transmission of heat through each section must give the quantity of heat transmitted by the wall, and these therefore must be equal to each other.

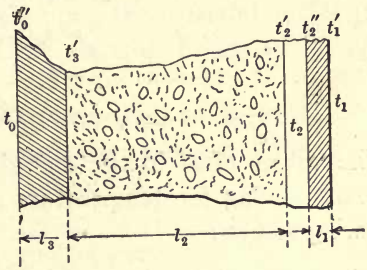


FIG. 96.—Wall Section.

The amount of heat conducted by any material per square foot of cross-section varies directly with the temperature difference and inversely with the length. This gives

$$H = \frac{C}{l}(t_1 - t_2), \quad \ldots\ldots (25)$$

where C is the constant of conduction for 1 ft. thickness in B.t.u. per square foot per degree, l is the thickness in feet and $t_1 - t_2$ is the difference of temperature. Using this for the wall shown in Fig. 96, the following results:

$$H = \frac{C_1}{l_1}(t'_1 - t''_2) = \frac{C_2}{l_2}(t'_2 - t'_3) = \frac{C_3}{l_3}(t'_3 - t''_0). \quad . \quad (26)$$

At the surface of any material there is to be found a temperature different from that of the contiguous space and it is this difference which determines the flow of heat at the surface.

At the surface the same formula

$$Q = \frac{C}{l}(t_1 - t_2)$$

holds, but since l is difficult to find, the quantity $\frac{C}{l}$ has been replaced by a and experiment is used to find the value of this for different materials and conditions of the surface. If a is the **coefficient of transmission** per square foot per hour per degree across this surface, this becomes at different surfaces:

$$\left.\begin{aligned} H &= a_1(t_1 - t'_1) \\ &= a_2(t''_2 - t_2) \\ &= a_3(t_2 - t'_2) \\ &= a_4(t''_0 - t_0) \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

The values of H in the sets above are all the same quantity, hence solving for temperature differences and adding, the following results:

$$H\left[\frac{1}{a_1} + \frac{l_1}{C_1} + \frac{1}{a_2} + \frac{1}{a_3} + \frac{l_2}{C_2} + \frac{l_3}{C_3} + \frac{1}{a_4}\right] = t_1 - t_0. \quad . \quad (28)$$

Now

$$K = \frac{H}{t_1 - t_0}.$$

Hence

$$K = \frac{1}{\frac{1}{a_1} + \frac{1}{a_2} + \frac{1}{a_3} + \frac{1}{a_4} + \frac{l_1}{C_1} + \frac{l_2}{C_2} + \frac{l_3}{C_3}}. \quad . \quad . \quad . \quad (29)$$

VALUES OF C

Air, still	0.03	Lead	20.00
Air in motion	0.09	Limestone	1.35
Asbestos paper	0.04	Lith	0.028
Blotting paper	0.04	Marble, fine	1.88
Brass	61.00	Mortar and plaster	0.46
Brickwork	0.46	Mineral wool	0.05
Building paper	0.08	Oak	0.13
Cement	0.40	Pine (along the grain)	0.11
Charcoal	0.03	Pine (across the grain)	0.06
Copper	202.00	Plaster of Paris	0.34
Coke	0.05	Sandstone	0.87
Cork, compressed	0.022	Sawdust	0.03
Cork, granulated	0.03	Shavings	0.05
Cotton	0.03	Slate	0.19
Feathers	0.040	Terra-cotta	0.54
Felt	0.02	Tin	35.60
Glass	0.54	Wool	0.03
Hair felt	0.026	Zinc	74.00

The values of the quantities a, as given from Grashof and Rietschel, are of the form

$$a=d+e+\frac{(42d+31e)T}{10{,}000}. \quad . \quad . \quad . \quad . \quad . \quad (30)$$

d and e are constants. d depends on the condition of the air around the surface and e depends upon the material. T is the temperature difference between the air and the surface at any point.

To determine the quantity T, a method of approximation is used until by practice one knows what to expect. The value of the term involving T,

$$\frac{42d+31e}{10{,}000}T,$$

is small, hence for a first approximation this term may be neglected and the value of the various a's may be found. These may then be used to find K.

$$K=\frac{1}{\Sigma\frac{1}{a_1}+\Sigma\frac{l}{C}}; \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

after this is known the following results:

$$K(t_1-t_0)=a_1(t_1-t'_1)=a_2(t_2-t'_2)$$
$$=a_1T_i=a_2T_0=\text{etc.},$$
$$T=t_1-t'_1 \quad \text{or} \quad t''_0-t_0.$$

These equations give the first approximations for T.

In this way after T is found as a first approximation, the value may be used to find a second value of a and then a new value for T. In this way two or three trials will lead to the correct result.

In any case the value of T is small and this is true particularly for thick walls or in cases in which t_1-t_0 is a small quantity.

Rietschel gives results used in practice for the value of T for masonry walls. These may be put into the form of an equation,

$$T=16.2-4.00l. \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

This may be used for masonry walls with air spaces where l is the sum of the various thicknesses of masonry, although the result is slightly too large in this case, as the quantity $K(t_1-t_0)$ is smaller than for a solid wall of the combined thickness.

For a single glass T is taken as $\frac{1}{2}(t_1-t_0)$, while for double windows $\frac{1}{4}(t_1-t_0)$ is taken at each surface, since glass is so thin there is practically no temperature drop in it, the main resistance being at the surface.

The value of T for wooden floors is given as $T=1.8°$ F. The values of d as given from Grashof are as follows:

VALUES OF d

Air at rest as in rooms or channels	0.82
Air with slow motion as over windows	1.03
Air with quick motion as outside of building	1.23

The values of the coefficient e are determined by Rietschel as follows:

VALUES OF e

Brass, polished	0.05	Polished sheet iron	0.092
Brickwork and masonry	0.74	Rusted iron	0.69
Cast iron, new	0.65	Sawdust	0.72
Cotton	0.75	Sheet iron	0.57
Charcoal	0.71	Tin	0.045
Copper	0.03	Water	1.07
Glass	0.60	Wet glass	1.09
Mortar and lime mortar	0.74	Wool	0.76
Paper	0.78	Zinc	0.049
Plaster of Paris	0.74	Wood	0.74

To explain the application of the above the wall given in Fig. 97 will be investigated. The wall is composed of 4 ins. of sandstone, 18 ins. of brick work, a 2-in. air space, 8 ins. of brick and 1 in. of plaster. Where sections of the wall actually

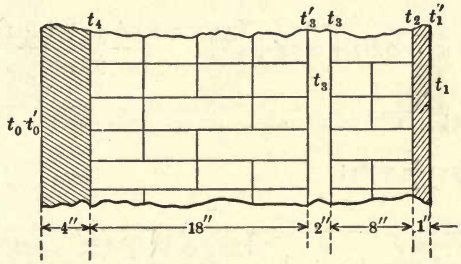


FIG. 97.—Wall Section.

come in contact, there is no surface resistance and the wall may be considered as solid except for differences in values of C for the various materials. When air can circulate it is not considered an insulator as the convection currents carry heat from the warm to the cold side. The value of air space is in the surface resistance. To find a the various values of T must be known; now T is given by the following:

$$T_1 = t_1 - t'_1;$$

$$T_3 = t''_3 - t_3;$$

$$T'_3 = t_3 - t'_3;$$

$$T_0 = t''_0 - t_0.$$

These quantities vary inversely with the different values of a, since

$$a_1 T_1 = a_2 T_3 = a'_2 T'_3 = a_4 T_0.$$

As the quantities a do not differ by great amounts these various values of T are considered as equal quantities in computing a.

T may then be found from the equation

$$T = 16.2 - 4.00l.$$

In this case the total thickness is 31 ins. and

$$T = 16.2 - 4 \times \frac{31}{12} = 6° \text{ F.};$$

$$a_4 = 1.23 + 0.74 + \frac{(42 \times 1.23 + 31 \times 0.74)6°}{10{,}000};$$

$$a_3 = a_2 = a_1 = 0.82 + 0.74 + \frac{(42 \times 0.82 + 31 \times 0.74)6°}{10{,}000};$$

$$a_4 = 2.01;$$

$$a_3 = 1.59 = a_2 = a_1;$$

$$K = \frac{1}{\frac{1}{2.01} + \frac{1}{1.58} + \frac{1}{1.58} + \frac{1}{1.58} + \frac{0.33}{0.87} + \frac{1.5}{0.46} + \frac{0.66}{0.46} + \frac{0.083}{0.46}}$$

$$= \frac{1}{0.497 + 0.633 + 0.633 + 0.633 + 0.379 + 3.26 + 1.435 + 0.180}$$

$$= \frac{1}{7.62} = 0.131.$$

The resistance of air channels is negligible because of the convection currents.

For **a floor or ceiling** as shown in A, Fig. 98, the method is quite the same. When the high temperature is at the top, however, there is no circulation in the air space between the plaster and the floor and the air acts as an insulating material.

When the high temperature is below or if an air space is in a vertical position, the circulation of the air transmits heat by convection and the air does not act as an insulating material.

In any case, however, there is a resistance at the surface between the air and the partition due to the drop T.

When the same constant K does not hold over a complete wall or floor owing to a change in the construction as occurs at studs in a partition or joists in a floor, the value of K for the whole surface is found thus:

$$K(F_1+F_2)(t_i-t_0)=K_1F_1(t_1-t_0)+K_2F_2(t_1-t_0)$$

$$K=\frac{K_1F_1+K_2F_2}{F_1+F_2}=\frac{\Sigma KF}{\Sigma F}. \quad . \quad . \quad . \quad . \quad (33)$$

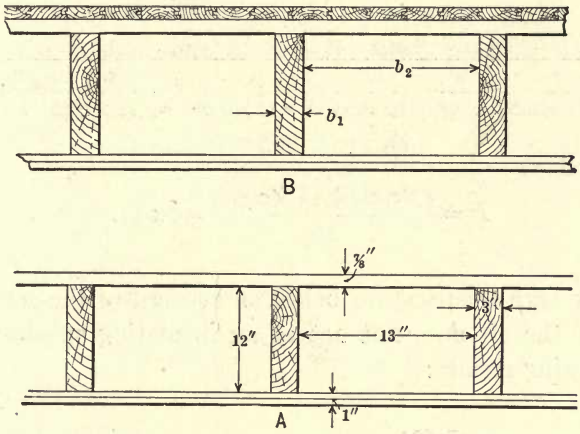


FIG. 98.—Floor Sections.

In most cases the areas F have a common dimension, so that the areas are proportional to the widths. If these are b_1 and b_2 there results (Fig. 98),

$$K=\frac{K_1b_1+K_2b_2}{b_1+b_2}. \quad . \quad . \quad . \quad . \quad . \quad (34)$$

The mean constant is not usually found for a wall in terms of glass and wall coefficient, as these are kept separate, but there is no reason why this could not be done, as happens with the coefficient for partitions with partition studs in the cases which follow.

With the high temperatures above the air acts as an insulating substance and the following for the floor, Fig. 98:

$$a_u = 0.82 + 0.74 + \frac{(42 \times 0.82 + 31 \times 0.74)1.8}{10{,}000} = 1.57;$$

at joists,

$$K_j = \frac{1}{\frac{1}{1.57} + \frac{13.25}{12 \times 0.06} + \frac{5}{8 \times 12 \times 0.46} + \frac{1}{1.57}} = 0.05$$

at space between joists,

$$K_a = \frac{1}{\frac{1}{1.57} + \frac{7}{12 \times 8 \times 0.06} + \frac{2}{1.57} + \frac{12}{12 \times 0.03} + \frac{3}{8 \times 12 \times 0.06} + \frac{5}{8 \times 12 \times 0.46} + \frac{1}{1.57}} = 0.027$$

Combined

$$K = \frac{3 \times 0.05 + 13 \times 0.027}{16} = 0.031.$$

With the high temperature below on account of the convection currents, the air does not act as an insulating substance and the following results:

$$a = 1.57;$$

$$K_j = 0.05;$$

$$K_a = \frac{1}{\frac{4}{1.57} + \frac{1.25}{12 \times 0.06} + \frac{5}{8 \times 12 \times 0.46}} = 0.22;$$

$$K = \frac{3 \times 0.05 + 13 \times 0.22}{16} = 0.19.$$

This method may be used for various walls and partitions. The following values have been computed by the author and these values compared with those given by Kinealy, Rietschel and others.

Values of a

For brick and plaster or masonry.

Outside $$a = 1.23 + 0.74 + \frac{43 \times 1.23 + 31 \times 0.74T}{10{,}000}$$

$$= 1.97 + 0.0075T$$

$$= 2.09 - 0.03l,$$

since $$T = 16.2 - 4l.$$

Inside $$a = 1.56 + 0.0057T$$

$$= 1.65 - 0.023l.$$

For wood and, approximately, paper, cotton, wool, coal and sawdust:

Outside $$a = 1.97 + 0.0075T = 1.98.$$

Inside $$a = 1.56 + 0.0057T = 1.57.$$

For glass:

Outside $$a = 1.83 + 0.007T$$

$$= 2.07\left(\frac{T = t_i - t_0}{2} = 35^\circ\right).$$

Inside with motion:

$$a = 1.63 + 0.006T$$

$$= 1.83(T = 35).$$

Inside without motion:

$$a = 1.42 + 0.005T;$$

$$= 1.59(T = 35).$$

Inside with motion and wet from condensation:

$$a = 2.11 + 0.008T$$

$$= 2.39.$$

For double windows:

Outside $a = 1.95 (T = \frac{1}{4} \times 70)$;

Center $a = 1.51$.

Inside, dry $a = 1.74$.

Pipe Covering. The use of pipe covering to prevent the conduction of heat from steam pipes or to brine pipes or vessels must be considered in this chapter. The discussion applies to covering on all circular bodies. The constants of this chapter may be used in this case. The transmission formula now becomes

$$Q = \pm \frac{c}{dr} 2\pi r L dt;$$

$$\frac{Q}{2\pi c L} \frac{dr}{r} = \pm dt;$$

$$\frac{Q}{2\pi c L} \log_e \frac{r_0}{r_1} = \pm (t_0 - t_1);$$

$$Q = \pm 2\pi r_1 L \left[\frac{c}{r_1 \log_e \frac{r_0}{r_1}} \right] \times (t_0 - t_1).$$

For **flat plates** of insulating material the expression to be used is

$$Q = \pm F \frac{c}{l} (t_0 - t_1).$$

For cork $c = 0.022$.

For

Q = heat per hour in B.t.u.;
c = B.t.u. per hour per degree for 1 ft. thickness;
r_0 = radius outside of covering in feet;
r_1 = radius of pipe in feet;
L = length of pipe in feet;
F = area of surface in square feet;
l = thickness of covering;
t_0 = temperature outside covering deg. F.;
t_1 = temperature inside of covering deg. F.

Mr. L. B. McMillan has recently given values for c for various temperature differences.

VALUES OF C

Kind of Covering.	Temperature Difference.							
	25	50	75	100	150	200	300	400
Johns-Manville asbestos sponge.	0.027	0.028	0.028	0.029	0.030	0.031	0.032	0.036
Nonpareil high pressure.....	0.033	0.033	0.033	0.033	0.034	0.034	0.035	0.037
Cary 85% magnesia	0.034	0.034	0.034	0.034	0.035	0.035	0.036	0.038
Johns-Manville magnesia.....	0.036	0.036	0.036	0.036	0.037	0.037	0.037	0.039
Carey carocel.....	0.029	0.030	0.031	0.032	0.033	0.035	0.039	0.044
Johns-Manville asbestocel.....	0.035	0.035	0.035	0.036	0.037	0.038	0.041	0.045
Johns-Manville aircell.....	0.038	0.038	0.039	0.040	0.041	0.043	0.047	0.054

The **insulating** of cold storage houses is accomplished by the use of **wooden walls with air spaces** as shown in Fig. 99, **brick walls with wooden backing** as shown in Fig. 100, **brick walls with air spaces** as shown in Fig. 101 and **brick walls lined with some non-conductor** as shown in Fig. 102. The main purpose in using these is to increase the heat resistance. The older storage houses were of wood and the method shown in Fig. 99 gave good satisfaction. The **use of paper or felt** coated with some substance to waterproof it keeps the sawdust and air space dry as well as making the wall air tight. **Sawdust or mineral wool** is used in the air space for the purpose of preventing air circulation. This is accomplished in air spaces by using horizontal strips which should be put at intervals between them. Fig. 100 shows a construction recommended by the Frick Company for warehouses. At times **cement, concrete or asphalt** is put on wooden floors as a wearing surface. Fig. 101 shows the **brick type of insulation** which is valuable although expensive. Where space is valuable some of the brick may be replaced by **cork board or by lith** as these have more resistance. The type shown in Fig. 102 illustrates such a protection. Two thicknesses of **cork board insulation with cement between** are used to get the necessary thickness, as these boards are usually made no greater

than 3 ins. in thickness. The cement is usually a tar, asphalt or some other waterproof binder. The surface is sometimes protected with a cement plaster of waterproof properties.

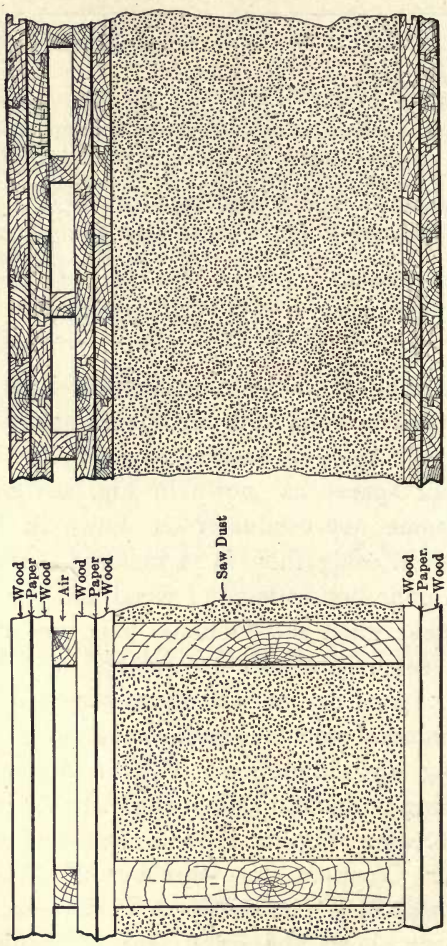


FIG. 99.—Wooden Wall with Sawdust Fill. (Elevation above, plan below.)

At times air spaces are introduced between the various thicknesses of boards as shown in Fig. 103, and in some cases the outer layer may be replaced by two of **lumber with paper between**. The combination used depends upon the peculiari-

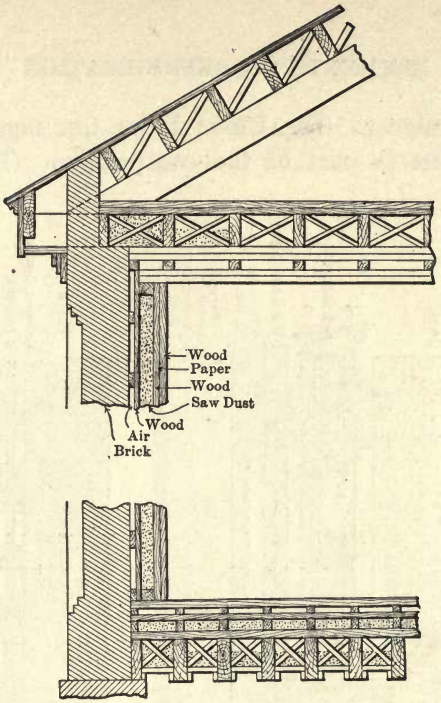


FIG. 100.—Brick Lined with Wood.

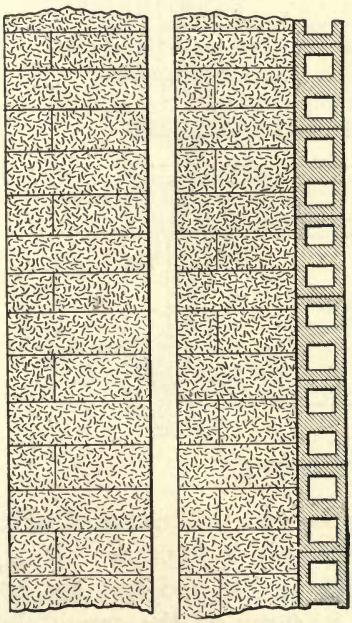


FIG. 101.—Brick Wall with Air Spaces and Tile Lining.

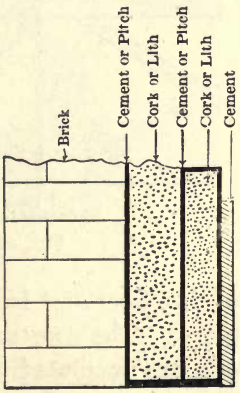


FIG. 102.—Wall with Lining of Cork or Lith.

ties of the designer. The Union Fibre Co. suggests the use of their **linofelt** as part of the construction. This is a felt

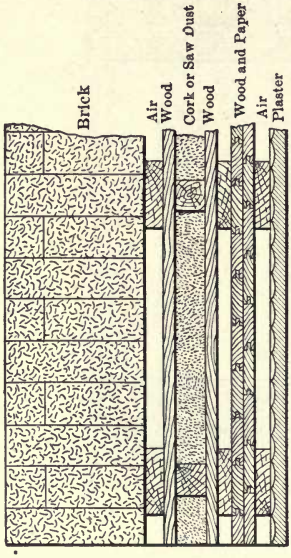


FIG. 103.—Brick Wall with Wood Lining.

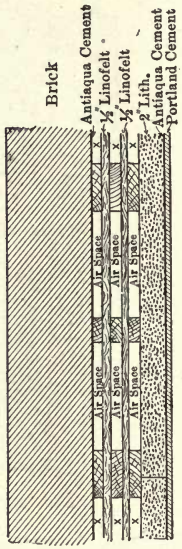


FIG. 104.—Use of Lith and Linofelt.

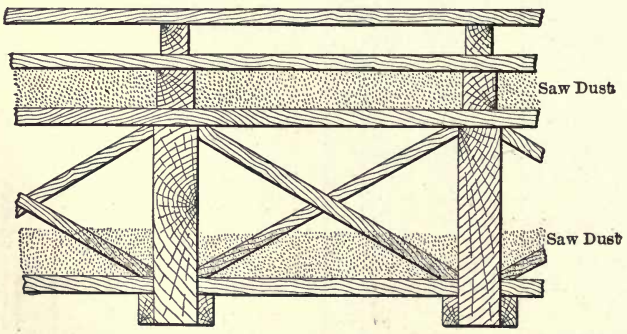


FIG. 105.—Floor Construction.

made of flax fiber and held between two thicknesses of waterproof paper. The construction is shown in Fig. 104.

Floors are insulated as shown in Fig. 100 and Fig. 105, when above the first floor, while for floors on the ground, Fig.

106 shows the method used. These are carefully drained and the endeavor is made to keep all moisture from the insulating

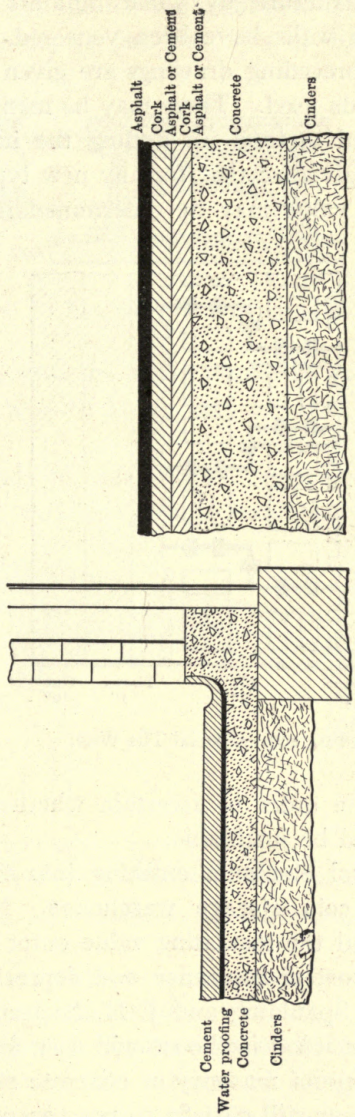


FIG. 106.—Ground Floor Construction.

material. Fig. 107 shows a form of wall using an interlocking and bonding section.

The **construction used in making grain bins** consisting of planks 2×10, 1×10 or 1×12 laid on the flat side, has been used for cold storage structure by some builders with success. In some cases such walls have been veneered with 4 ins. of brick. All of the preceding drawings are given to show some of the many methods used. There may be many changes suggested. The general method for **finding the insulating value of a wall** has been given so that for any new type of construction the insulating value may be determined before the con-

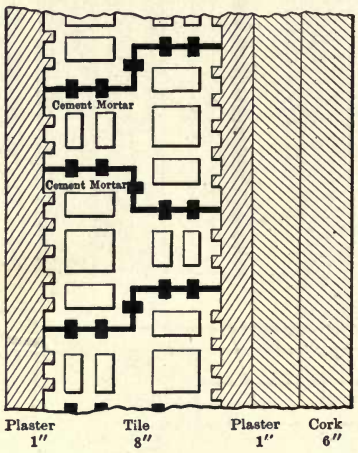


FIG. 107.—Special Tile Wall.

struction is made, in order to ascertain whether or not additional expense would be justifiable.

There are several **elements entering into the problem of construction of a cold storage warehouse.** Not only does the **original cost** and the **insulating value** enter into the problem, but also the **cost of insurance and depreciation** must be considered. R. E. Spaulding and J. H. Nielson have pointed out that although a wood ice-house will cost $2.00 per ton of capacity, and a fireproof masonry or concrete structure of the same insulating power will cost $2.50 per ton of capacity, the latter costs less to operate because the depreciation must be figured at 10% for the wood, and at 3% for the fireproof

structure, building insurance $5.00 per hundred on 80% of the wooden building and 40 cents on 80% of the fireproof building while for the ice the insurance is $5.00 for wood and 40 cents for concrete per 100 tons of ice. Considering the

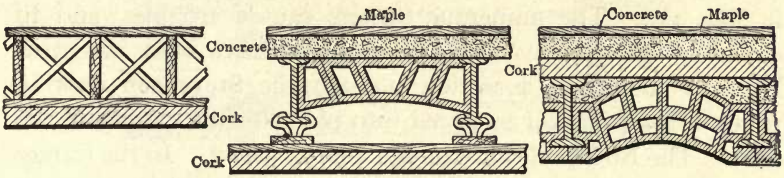


FIG. 108.—Floor Construction.

interest at 5% with the above items the yearly cost is 43 cents for the cheap wooden house, and 21 cents per ton in the fireproof house.

The methods of this chapter have been used to compute the values for various types of insulation and the results are

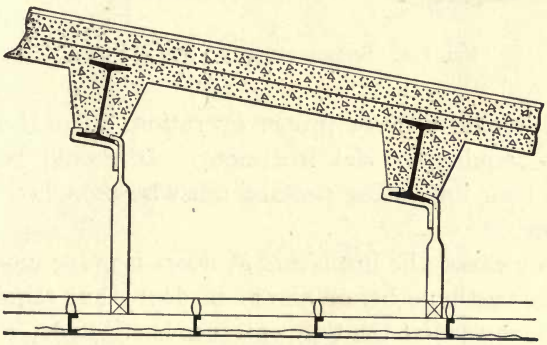


FIG. 109.—Reinforced Concrete Roof and Ceiling.

given on p. 211. These values may be used, if desired, to make preliminary calculations.

Fig. 108 shows a **construction of floors** using arches while Fig. 109 illustrates the method of hanging a ceiling on an inclined reinforced concrete roof to form an air space or to give a level ceiling.

The **construction of doors** is an important question in the

operation of a warehouse. Not only must these be non-conducting, but they should be air tight, because the temperature difference might set up a strong circulation of air through cracks, and this must be avoided.

To prevent this, **leakage doors** were originally made as shown in Fig. 110. The numerous corners caused troubles, and to do away with them other arrangements have been invented. Fig. 111 illustrates a section through the **Stevenson door** in which a hemp gasket is forced into place by the closing of the door. The **No Equal cold-storage door** is shown. In the former the soft gasket projects from the door flange while in the latter the hair felt which is inclosed in a ring of canvas or rubber is placed in two grooves beneath a gasket of rubber or leather in the concave quadrant corner of the door. The jamb is rounded, removing all sharp corners which would be bruised

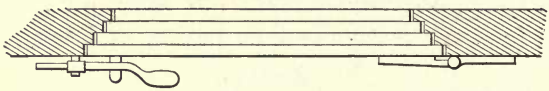


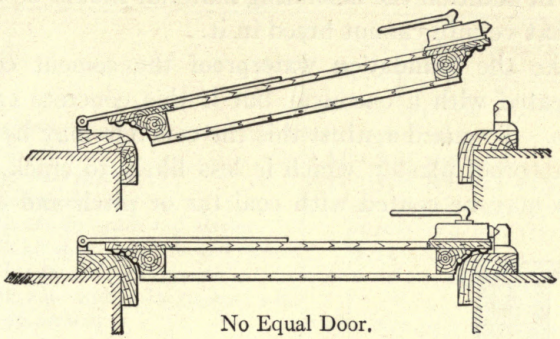
FIG. 110.—Section of Early Form of Door.

and which would prevent proper operation. The threshold of the doors requires special treatment. It should be beveled off to the floor line. The packing must be tight here to make a proper fit.

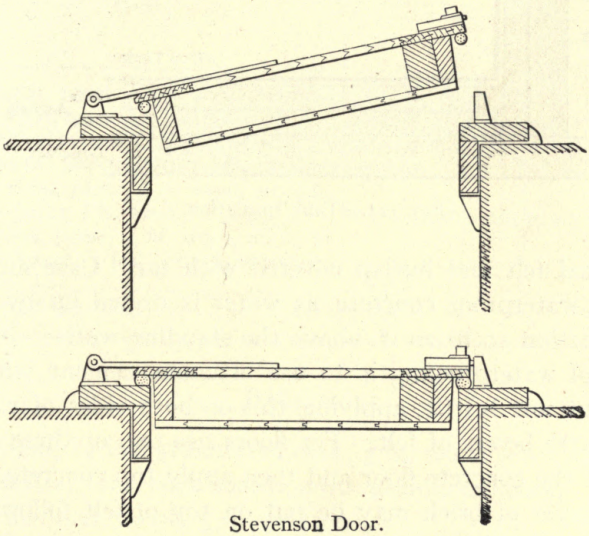
In many cases, the insulation at doors is made more perfect by using a vestibule before the main door, thus requiring two doors to be opened at the time entrance is effected.

The **insulation of ice tanks** is shown in Fig. 112. The method of construction and computation is the same as that used above. The tank may rest on several layers of cork board or on wooden sleepers and the sides may be insulated with granulated cork. The heat loss must be cut down to a low value.

In all of the above methods of insulation care must be taken to **prevent moisture from entering** the insulation, as the value is decreased when this becomes wet and the wood or material



No Equal Door.



Stevenson Door.

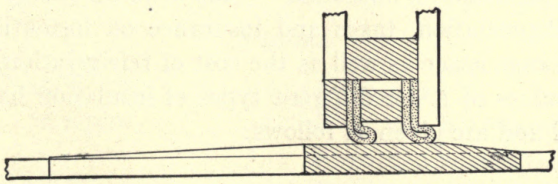


FIG. 111.—Arrangements of Doors.

may rot. In addition the insulating material should be of such a nature that vermin cannot breed in it.

To make the **foundation waterproof** the cement concrete may be treated with a chemical, but if this concrete cracks a leak occurs. To guard against this the concrete may be coated with a waterproof plaster, which is less likely to crack, or the foundation may be coated with coal tar or pitch and covered

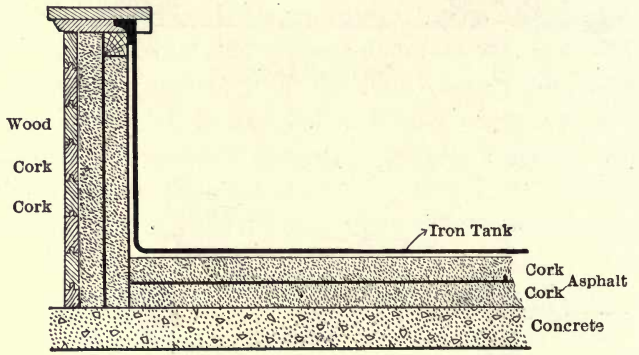


FIG. 112.—Tank Insulation.

with tarred felt and burlap covered with tar. Care must be taken to waterproof concrete, as water is drawn up by capillary attraction 10 to 20 ft. above the standing water. A good method of waterproofing is to use bitumen cement, which is strong but not brittle, applying this on both sides of each of two or three layers of felt. For floors use two or three layers on top of the concrete floor and then apply top concrete. For roofs, a layer of brick may be put on top of felt followed by 6 ins. of earth with grass.

One of the chief points to consider is to make the yearly cost of refrigeration a minimum. This includes yearly cost for interest, depreciation, taxes and insurance on insulation, with cost of storage space as well as the cost of refrigeration.

The values of K for different types of insulation have been computed and are given as follows:

VALUES OF K

Walls.	Total Thickness of Brick Masonry.							
	4″	8″	12″	16″	20″	24″	28″	32″
Solid brick	0.55	0.39	0.31	0.25	0.21	0.18	0.16	0.15
Solid brick with plaster	0.51	0.37	0.29	0.24	0.21	0.18	0.16	0.15
Brick with one air space	....	0.27	0.22	0.19	0.17	0.15	0.13	0.12
Brick with one air space and plaster	....	0.26	0.22	0.19	0.17	0.15	0.13	0.12
Brick with air space, 4-in. tile and plaster	....	0.14	0.13	0.12	0.11	0.10	0.09	0.09
Brick with 3-in. cork and plaster	0.07	0.07	0.07	0.06	0.06	0.06	0.06	0.06
Brick with 2-in. cork, ½-in. cement, 2-in. cork, ½-in. cement	0.06	0.06	0.05	0.05	0.05	0.05	0.05	0.04

Walls.	Sawdust Thickness.					
	0″	6″	12″	18″	24″	30″
⅞ in. wood, paper, ⅞ in. wood, sawdust, ⅞ in. wood, paper, ⅞ in. wood	0.167	0.044	0.025	0.018	0.014	0.011
Same with shavings in place of sawdust	0.167	0.062	0.039	0.028	0.022	0.018
⅞ in. wood, paper, ⅞ in. wood, sawdust, ⅞ in. wood, paper, ⅞ in. wood, air, ⅞ in. wood, paper, ⅞ in. wood. Fig. 99	0.102	0.038	0.023	0.017	0.013	0.011
⅞ in. wood, paper, ⅞ in. wood, air, ⅞ in. wood, paper, 3 ins. cork, ½ in. cement plaster	0.057					
Same with 6 ins. cork	0.036					

PARTITIONS

⅞ in. wood, 12 ins. granulated cork, ⅞ in. wood. . . . . $K=0.024$

½ in. plaster, 3 ins. cork boards, 4 ins. granulated cork, 3 ins. cork board, ½ in. plaster. . . . . $K=0.027$

Tile partitions plastered single. . . . . $K=0.30$

Tile partitions plastered double. . . . . $K=0.21$

FLOORS

Fig. 108, 1st figure. . . . . $K=0.022$

2d figure. . . . . $K=0.062$ heated room below

$K=0.030$ heated room above

3d figure. . . . . $K=0.060$

12 ins. concrete, 2 to 3 ins. cork boards, 2 ins. cement $K = 0.038$
Glass, single thickness ..... $K = 1.06$
Glass, air, glass ..... $K = 0.41$
Glass, air, glass, air, glass ..... $K = 0.26$
6 glass and 5 air layers ..... $K = 0.12$

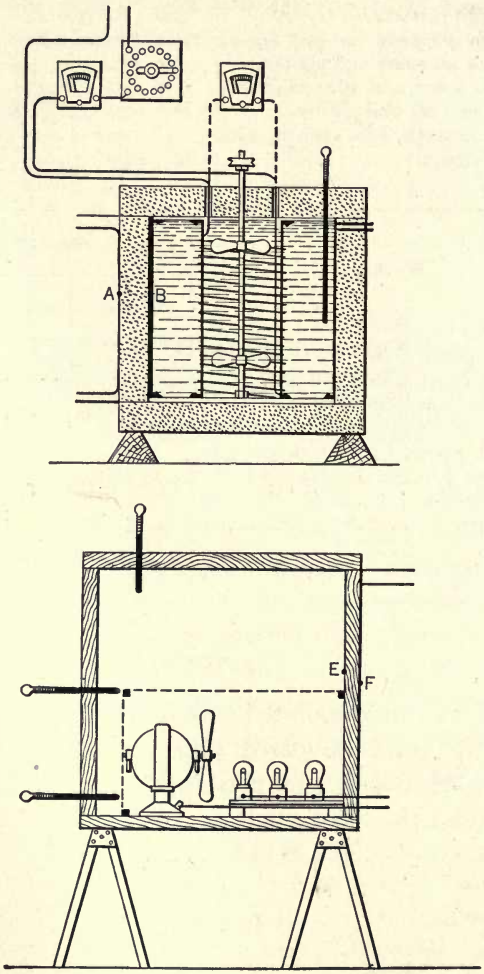


FIG. 113.—Norton's Method of Finding K.

The **value of K for cork board** has been found by Prof. Norton in several ways. In one case he built a cubical box

of the cork to be tested, and placed a piece of ice within. By weighing the amount of water from the ice the heat carried in, was found. In a second test a fan was placed within the box and electric lamps or resistances were used to produce heat and by circulating the air the temperature was made uniform; then by measuring the energy to hold the box at some temperature above the room temperature, the heat loss per degree was found. To get the area of the box surface, the surface of a cube at the mean thickness of the cork was computed. In addition to this the heat added to warm oil circulated in a tin lining on the inside of the box was found electrically and reduced to B.t.u. per square foot per twenty-four hours. Norton also placed a wire grid between two thicknesses of cork board. After allowing for heat losses at the edge by keeping grid at a certain state the heat loss in electrical energy per square foot per degree per hour was found. The mean value suggested by him for cork is $K=0.022$. These methods have been used by German experimenters and others for the determination of K for various substances.

There are other **elements entering into the heat supply** of refrigerating plants. Any air leakage or ventilating supply must be cooled off. For M lbs. of air per second, the heat per hour will be:

$$Q=3600Mc_p(t_0-t_i)$$

$$=60\frac{pV}{BT}c_p(t_0-t_i);$$

M = weight of air per second;
c_p = specific heat of air = 0.24;
t_0 = temperature outside in deg. F.;
t_i = temperature inside in deg. F.
p = pressure in pounds per square inch;
V = volume per minute in cubic feet at p lbs. pressure and absolute temperature T;
$B=53.35$.

The **heat produced by persons** is given by the following results by Benedict in the table below:

Adult at rest, asleep	258 B.t.u. per hour
sitting	396
Adult at light work	670
Adult at moderate work	1150
Adult at severe work	1780

For children an allowance of 300 B.t.u. per hour may be made.

Heat of Machines and Lights:

For electric lights	1 watt-hour	3.41 B.t.u.
For power	1 K.W.-hr	3410
	1 H.P.-hr	2546

For gas where used:

1 cu.ft. illuminating gas	700 B.t.u.
1 cu.ft. natural gas	1000

1 Welsbach burner uses 3 cu.ft. of gas per hour.

1 fish-tail burner uses 5 cu.ft. of gas per hour

If substances are chilled the following **specific heats** and **constants** in table on page 215 are used.

These values have been obtained by reference to various authors and are collected here in a separate table.

See Storage Rate Guide for rules relating to charges for storage, rules for labor charges, liability, etc.

Having the amount of goods put in a storage room the heat per hour to cool this **refrigeration in tons** is:

$$\text{Tons} = \frac{Q}{199.2 \times 60} = \frac{Mc(t_a - t_r) + Ml_t}{h \times 199.2 \times 60}. \quad . \quad . \quad . \quad (35)$$

M = weight of goods including weight of container;

c = specific heat of substance;

t_a = temperature of outside air or temperature of goods put in storage;

t_r = temperature of storeroom or temperature of goods after storage;

l_t = heat of fusion if goods are frozen;

h = hours required to cool goods;

Q = B.t.u. per hour.

COLD STORAGE DATA

Substance.	Temp. Deg. F.	Specific Heat. Before Freez.	Specific Heat. After Freez.	Latent Heat of Fusion.	Time of Storage.	Cost of 1 mo. Storage. Cents.	Cost of Each Successive mo. Cents.	Unit of Storage.
Apples	30–35	0.92	....	....	6 mo.	20	15	Barrel
Bananas	34–40	....	....	....	3 mo.			
Beans, green	36–40	0.91	....	....	....	10	10	Bu. basket
Beans, dried	40–45	0.90	....	....	....	15	10	100 lbs.
Beef, fresh	30–38	0.70	0.38	90	3 mo.	¼ to ⅛	¼ to ⅛	1 lb.
Beef, salt	40–45	0.60	....	....	....	⅛	⅛	1 lb.
Beer	30–36	0.90	....	....	....	12½	10	Half barrel
Berries	36–40	0.91	....	....	short	1 to ½	1 to ½	Quart
Butter	15–20	0.60	0.84	84	5 mo.	¼	⅛	1 lb.
Cabbage	32–36	0.93	0.48	129	....	20	15	100 lbs.
Cantaloupes	34–36	0.92	....	....	....	10	7½	Box or crate
Cherries, fresh	36–40	0.92	....	....	3 wks.	15	12	½ bu. basket
Cherries, dried	40–45	0.84	....	....	....	12½	12½	100 lbs.
Cheese	32–36	0.64	....	....	4 mo.	15	10	100 lbs.
Celery	34	0.91	....	....	....	25	25	Large crate
Cider	30	0.90	....	....	....	25	20	Barrel
Cigars	40–45	....	....	....	....	5	5	Per cu.ft. at $5 val.
Dates	40–45	0.84	....	....	....	12½	12½	100 lbs.
Eggs	30–31	0.76	0.40	100	6 mo.	10	7½	30 doz. case, 55 lbs
Fish, fresh	0–5	0.75	0.40	100	8 mo.	¾	¼	1 lb. glazing
Fish, dried	35–40	0.58	....	....	....	20	15	100 lbs.
Fruit, dried	35–40	0.84	....	....	....	12½	12½	100 lbs.
Furs, coats	25–30	....	....	....	....	$1.50 to	2.00	per season each
Furs, rugs	25–30	....	....	....	....	$ .60 to	.75	per season each
Furs, uncured	30–35	....	....	....	....	$ .60 to	.75	per season 1 cu.ft.
Game	15–20	0.80	0.40	105	....	¼	¼	1 lb.
Grapes	32–36	0.92	....	....	2 mo.	5	4	10 lb. basket
Grape fruit	32–40	0.92	....	....	3 mo.	10	7½	Box
Lemons	32–50	0.92	....	....	3 mo.	10	7½	Box
Milk	32–36	0.90	0.47	124	....	12½	12½	40 qt. can
Mutton	30–32	0.67	0.81	....	5 mo.	⅛	⅛	1 lb.
Onions	32–36	0.91	....	....	6 mo.	15	15	2 bu. sacks
Oranges	32–50	0.92	....	....	3 mo.	10	7½	Box 100 lbs.
Oysters, bulk	30–34	0.84	0.44	114	....	10	10	Tub
Oysters, shell	36–42	....	....	....	....	25	25	Sack
Peaches	32–36	0.92	....	....	1 mo.	5	4	½ bu. basket
Pears	30–36	0.92	....	....	2 mo.	5	4	½ bu. basket
Pork, fresh	30–32	0.50	0.30	90	1 mo.	⅛	⅛	1 lb.
Pork, cured	40–45	....	....	....	....	⅛	⅛	1 lb.
Potatoes	34–38	0.80	0.42	105	6 mo.	25	25	Barrel 2½ bu.
Poultry	15–20	0.80	0.40	102	3 mo.	¼	¼	1 lb.
Sausage, fresh	36–40	0.70	....	....	....	⅛	⅛	1 lb.
Sausage, smoked	40–45	0.60	....	....	....	⅛	⅛	1 lb.
Strawberries	36–40	0.92	....	....	short	½	½	1 qt. box.
Vegetables, bbl.	36–40	0.91	....	....	2 to 4 weeks	25	25	1 barrel
" crate.	36–40	0.91	....	....	2 to 4 weeks	12	12	1 bu. crate
Watermelons	34–36	0.92	....	....	....	5	5	1 melon
Wines	40–45	0.90	....	....	....	35	35	1 barrel

By use of Eq. (35) the amount of refrigeration to cool the goods and freeze them may be computed. It is well

to note that the time required to do this is an important factor. If the time is short the amount of refrigeration is large. This amount of refrigeration for this reason may be much larger than that required to care for the heat loss from the room.

CHAPTER VI

COLD STORAGE

THE purpose of cold storage is to prevent the development of life which would cause decay of living tissue; it is also used to prevent the development of living organisms. It is used not only for the storage of foodstuffs, but for the storage of furs, trees, flowers and other articles which require a low temperature for their proper keeping.

The principal application of cold storage is to the storage of food products. In 1905 W. T. Robinson stated that there were over $200,000,000 worth of products stored, divided between (*a*) living substances, such as eggs and fruit, requiring a moderate temperature and (*b*) non-living, as meats, butter and cheese, requiring a low temperature. In 1909 the value of goods passing through cold storage amounted to $2,585,000,000, ranging from $25,000,000 in fish to $1,500,000,000 in meats, the meat annually chilled alone amounting to 20,000,000,000 lbs. There were 160,000,000 cubic feet of storage space exclusive of breweries, packing houses, creameries and stores.

These goods are stored for various lengths of time. Meats may be frozen and then stored for a long time. There is some improvement in quality at first and although with lengthy storage there is no deterioration in the meat, the flavor is lost and for that reason long holdings are not good. Poultry may be frozen and for a certain length of time there is an improvement in quality. Eggs may be held for long periods and except for a loss of weight there is no ill effect. Cheese improves as it ripens in cold storage but after ripening there is no improvement. Butter suffers slightly in taste on long storage. Apples and pears are improved by holding, as certain

chemical changes take place, while strawberries and peaches lose their flavor rapidly. These various articles require special temperatures for their storage and hence there must be special rooms in warehouses for each article.

The value of cold storage is to equalize the supply of foodstuffs and make it possible to have certain foods during the whole year. The consumers claim that goods are held until the off-season and then exorbitant prices are asked, while the cold-storage men claim that prices are reduced by the ample supply which exists in the off-season. Formerly one of the great evils of the business was the lengthy storage of articles for times of high prices, hence **laws** have been made in many States to correct the evils of cold storage of foodstuffs which have hampered the business and brought about other evils. The United States Government is planning a **national cold-storage law** to cover interstate business and business in the District of Columbia.

The **cold-storage bills** define **cold storage** to be any receptacle where for periods longer than ten days food products are kept at 40° F. and under. There is usually a time limit for most substances; this varies from nine to twelve months. The materials stored must have the dates of receipt and delivery by the warehouse stamped on them and no restorage is permitted in some States. No cold-storage goods with dates erased may be sold. When eggs and butter are stored they must be sold as refrigerated articles and signs should state this. This refers to eggs after thirty days. In some States there are fines for the first two offenses and fine and imprisonment for the third offense. An important feature covered by U. S. Senate Bill 136 was the requirement that no food could be placed in cold storage unless in a sanitary condition. The condition when received and previous history of an article to be stored is as important as the storage. The Senate bill prohibits the manipulation of cold storage goods to resemble fresh goods and frozen articles must be sold in that condition.

An investigation by the U. S. Department of Agriculture showed that in three months the various percentages of stored

goods delivered from storage in certain warehouses expressed as a percentage of goods received at the beginning of the period were as follows:

Beef	71.2%
Mutton	28.8
Pork	95.2
Poultry	75.7
Butter	40.2
Eggs	14.3
Fish	35.5

And in seven months the amounts used were:

Beef	99 %
Mutton	99.3
Pork	99.9
Poultry	96.1
Butter	88.4
Eggs	75.8
Fish	64.9

The average months of storage were as follows:

	1st Half Year Months	2d Half Year Months
Beef	2.6	1.8
Mutton	4.8	3.0
Pork	0.8	1.0
Poultry	2.6	2.4
Butter	4.5	4.0
Eggs	6.1	1.7
Fish	6.8	6.7

This investigation shows that goods do not remain in storage for a long time.

The goods stored are handled in special ways and these will now be discussed together with certain data to be used in designing cold-storage warehouses.

Eggs. It has been stated that of the 3 billion dozen eggs produced in the United States yearly, 240 million dozen, or one-twelfth, are put into cold storage. Eggs are usually placed in **cases** containing 30 doz. These cases weigh about 50 to 55 lbs. and are usually stored in tiers five or six high with slats between cases to give a chance for air circulation and the removal of heat. These cases are $12 \times 13 \times 25$ ins. and occupy $2\frac{1}{4}$ cu.ft. of space. The eggs lose weight on storage, about 7% being lost in five or six months. If the air in the storeroom is too dry there is considerable loss of weight, while damp air will cause a fungous growth on the eggs. In many cold-storage warehouses there is no forced circulation, the ice on the pipes keeping the air a proper humidity. Should the air become too moist it may be dried by putting calcium chloride trays on top of the refrigerating coils and draining off the solution formed. This salt may be regained by evaporating the water. 80% relative humidity has been found to give good results.

The eggs are held at 30 or 31° F. and as they absorb odors they should be placed in rooms containing eggs only. They are placed in storage in April, May and June and are usually kept for about nine months. They have been kept for twenty-three months and except for a shrinkage of 25% they were not affected by storage. The cost of this storage is 10 cents per case for the first month and $7\frac{1}{2}$ cents for subsequent months; 40 to 45 cents would be the charge for the season. At 20 cents per dozen for the original eggs the item of 30 cents for the case, 30 cents for freight, 40 cents for storage, 25 cents for interest and insurance, and 40 cents for buying, packing and grading makes the price per dozen 25.5 cents, leaving about 5 to 10 cents per dozen margin for the owner, wholesaler and retailer. January is considered the end of the season. The eggs should not be washed when put in storage, as this spoils the appearance. At times they are **candled** before storage, although this is not done regularly. **Candling** consists of holding an egg in front of an opening in a metal screen, Fig. 114, within which is an electric light (originally a candle). If the egg is not good

a dark center due to the thickening of the yolk will be noted. A good egg will appear practically uniform in texture, the light shining through the egg. Candlers become very expert and this work is done rapidly. Candling is often done when eggs are taken from cold storage. Good cracked eggs are broken open and the meat placed in cans holding about 50 lbs., which are sealed and frozen. These are used by bakers. It is necessary to use these soon after thawing.

Uncracked eggs are broken and canned to reduce the cost

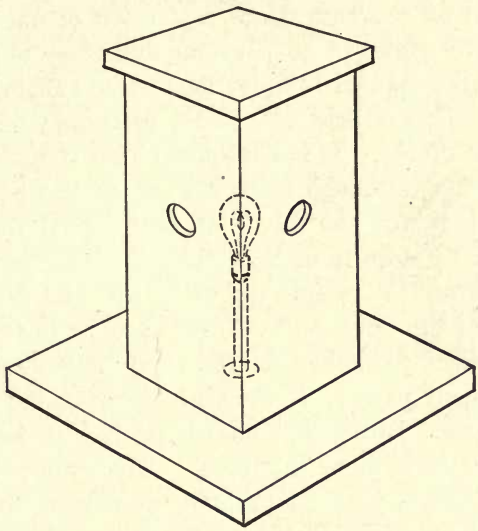


FIG. 114.—Candling Box.

of shipment. In Sedalia, Mo., a large plant is installed for cracking eggs. Here $10\frac{1}{2}$ millions of eggs are cracked during a season under highly sanitary conditions to prevent contamination of the egg meat. The eggs are broken on a knife and the whites are separated from the yolks, the latter being well mixed before sealing the can. This holds 30 lbs. The canned cracked eggs are frozen and shipped to bakers for consumption. It happens that the output of this plant is used by one baker alone.

The shipment of eggs from China is increasing. The U. S.

Commerce reports from Shanghai, China, state that the shipment of eggs amounts to 800 to 1000 tons per month, 1 ton being 40 cu.ft. This is about 400,000 doz. per month.

The storerooms for eggs and for all other storage should be kept clean and should be whitewashed about once a year.

The **whitewash** used to sweeten rooms can be made with a bushel of lime slaked in boiling water with a peck of salt and enough water to make a thin paste. To each 12-qt. pail of this add a handful of Portland cement and a teaspoonful of ultramarine blue to overcome a yellow discoloration. To prevent **dust on concrete floors**, a solution of one part sodium silicate (water glass) of 40° Beaumé and three to four parts of water has been applied to a dry floor after washing.

Butter. This is held at 33° F., or it may be frozen at a temperature of 15°. It is considered that it loses flavor with time of storage, although instances are given where the buyer has not questioned the flavor of butter held for two years. It is better to store it in bulk than in small packages. The ordinary butter tub weighs 50 to 60 lbs. and occupies about 2 cu.ft. The tub must be sweet and clean and carefully closed. It is sometimes paraffined on the inside and sometimes lined with parchment paper. If the air is damp mould will form on the parchment. Butter will absorb odors and for that reason it should be placed alone in a room. The amount in storage in the United States in 1909 probably amounted to 100,000,000 lbs., while the total production of the country is about eighteen times this amount.

The **temperature** does not seem to help in preserving the flavor. On judging cold storage butter there was little difference in the total points of butter at −10° F. and 10° F., but butter at 10° F. was much better than butter at 32° F. It depends on the kind of butter to a large extent. According to Gray, high salt content and hermetical sealing are not advantageous in preserving flavor. A common method of storage is to chill the butter at first to 0° F. and then allow the temperature to rise to 16 or 20°. Oleomargarine and such products may be held at about 20° F.

The **temperature carried is a question of economy;** the cost of refrigeration is placed against the saving in value of flavor. The limit of time of storage is about eleven months. The **house** should be **sweetened** with whitewash and a washing of $\frac{1}{1000}$ bichloride of mercury in water once a year.

A mixture of **ice and salt** is sometimes used for the cooling of these rooms. Cooper reports a butter-freezing room in Kentucky where 1700 cu.ft. is held at $10\frac{1}{2}°$ F. during August by the use of 507 lbs. of ice and 109 lbs. of salt per day. The egg room of 3560 cu.ft. is held at 30° F. by 790 lbs. of ice and 132 lbs. of salt. At this place such a method was considered best with ice at $2.50 per ton and salt at $7.00 per ton.

Cheese. Cheese weighs 60 lbs. to the box and occupies about 2 cu.ft. It is stored at about 32° F., although 36 to 40° is used. The U. S. Government tests were made at 31 to 32° F. Until it is thoroughly ripened this storage improves the cheese. Beyond that time it is not improved. The cold storage will check ripening and so keep the cheese. It is really to control the ripening that refrigeration is used. To prevent loss of weight it is customary to coat the cheese with paraffin. It should not be frozen.

Meat. Meat is improved by exposure to cold for a short time if kept at 25 to 28° F., but after about three weeks it gradually loses its flavor, although the meat is preserved. The fresh meat from the slaughter-house is placed in chilling rooms and it is cooled to the temperature of the main storehouse. In this way the chill room is equipped with excessive coil-cooling surface so as to remove the heat at the proper rate. Siebel states that 80 B.t.u. of refrigeration per twenty-four hours is required for every cubic foot of chill room. He states that 1 ft. of 2-in. direct-expansion pipe or 2 ft. of 2-in. brine pipe will care for 14 cu.ft. of chill room. If the meat is to be frozen for storage it is placed in a room at 10° F. and an allowance of 200 B.t.u. per twenty-four hours per cubic foot is made by Siebel and one-half the previous allowance per foot of pipe is used. This freezing is resorted to for shipment and for storage. This partially destroys the flavor. If thawed

slowly the flavor is not lost. The freezing should be done slowly and meat should not be stored in such large piles that the heat cannot be removed from the center. It should be held so that heat may be taken from all parts. If this is too rapid, the outer layer freezes before the inner part, and this leads to certain decay at the center of the meat. The amount of refrigeration may be computed from the specific heats and heats of fusion given in Chapter V when the weights are known. The weight will vary, but the following averages may be used:

Beef (two halves)	750 lbs.
Calves	90
Sheep	75
Hogs	250

The time of cold storage of meat may be at least six months if there is no chance of thawing.

Poultry. The storage of poultry has been a practice for some years. The poultry is frozen and kept in this condition. Dr. M. E. Pennington has investigated the matter of storage and care of poultry for shipment for the U. S. Government. She points out that the preparation of the fowl for storage is as important as storage itself. The chickens should be starved for twenty-four hours before killing to remove the putrid matter in the intestines, then the blood should be removed from the tissues after killing and the picking should be done without breaking the skin. This should be done dry and not after scalding the carcass. The carcass and especially the feet should be cleaned and prompt storage after chilling should be resorted to. With care of this kind the poultry is good after three weeks even if not frozen. The chill room is held at from 33 to 38° and the packing at from 30 to 32°. Certain State laws allow ten months storage of poultry. This is accomplished by freezing. In all cases the entrails are undrawn from the carcass. The poultry is usually placed in small boxes or barrels. The packages should be small so that air can reach all parts. The boxes should not be piled until after the poultry is frozen.

Milk. Milk, if free from the germs of fermentation, will keep indefinitely, but this condition is difficult to attain, and for that reason the growth of the germs is prevented by lowering the temperature. Of course the pasteurizing of the milk by warming it to a temperature of 180° F. will kill the bacteria and not scorch the milk. This is followed by rapid cooling. Milk should be cooled as soon as possible after being drawn from the cow. The temperature at which it is held is about 40° F., for if frozen there is a formation or separation of flocculent particles of albumen or casein compounds which do not redissolve readily on thawing. Fat globules or lumps are formed also. The cooling of the milk is accomplished in special block-tin coolers arranged so that they may be thoroughly cleaned. The milk passes over the outside and the refrigerant on the inside. One of these coolers is shown in Fig. 115. The Creamery Package Co. allows 20 sq.ft. of surface in these coolers per 1000 lbs. of cream or milk per hour, making the cooler of $1\frac{1}{2}$ or 2-in. copper or steel tubes, tinned on the outside. The trough at top or bottom is made of tinned copper. Other coolers are made with a hollow screw, which rotates in the ripening box, while the screw is furnished with a cooling solution. By having a hot supply in the screw the box acts as a pasteurizer and on following this with a cool solution the milk is cooled and made uniform by the turning of the screw.

Cream. In the storage of cream a low temperature is necessarily combined with clean storage vessels. This cream is used largely for butter-making and in many of the eight thousand creameries of the United States refrigeration is not employed, resulting in a poorer quality of butter, as cream is often held for some time by farmers before shipment to the creamery is made. The separation of the cream is carried on by the de Laval separator at the dairy or creamery and this may be done best at about 160° F. The cream after being cooled to 50° is stored and finally allowed to ripen at 70° F.

Fish. Fish is usually frozen and coated with ice before shipment and storage. This method is highly developed in the Northwest, 1000 carloads of halibut being shipped yearly from

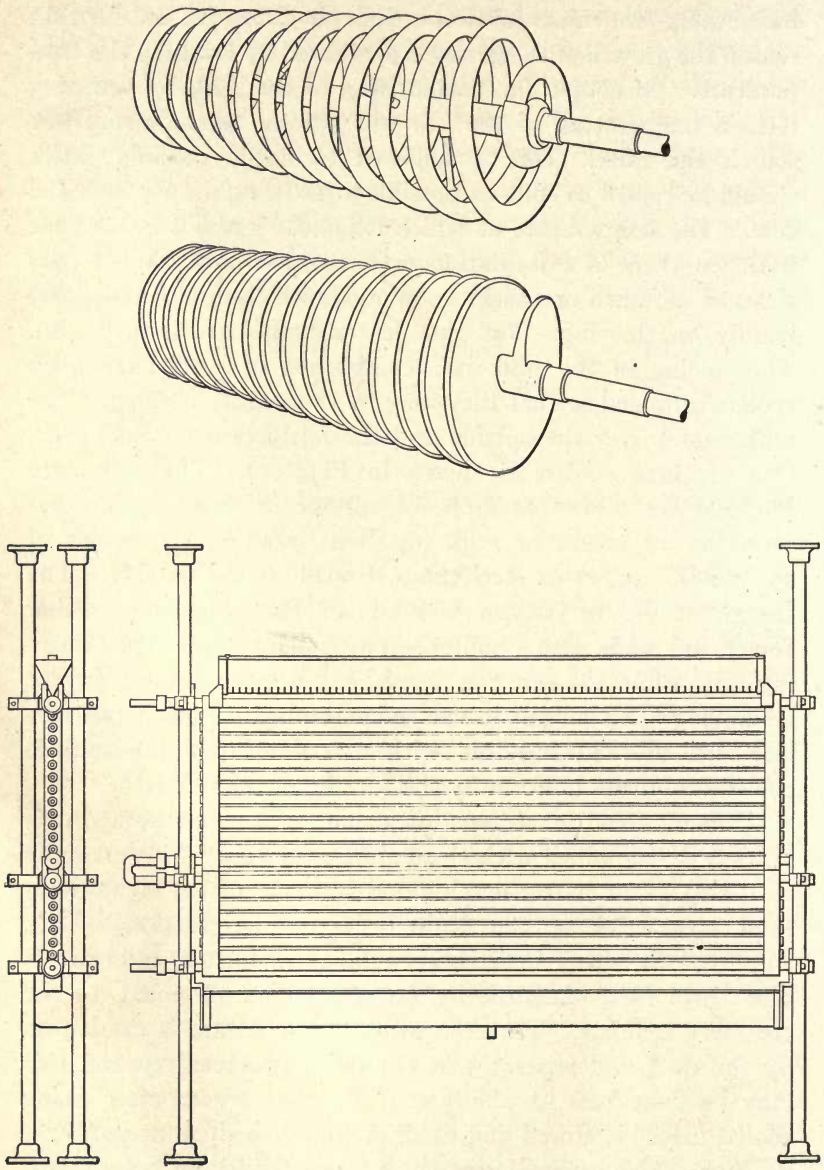


FIG. 115.—Spiral Coil, Disc Coil and Tubular Cooler of Creamery Package Co.

Vancouver. The fish are decapitated, cleaned, washed and placed in sharp freezers, the refrigerating coils acting as shelves. Certain of these rooms are equipped with two sets of eight shelves made up of 1-in. extra heavy pipe 37 ft. long. These pipes are supplied with liquid ammonia for direct-expansion, and by keeping liquid in each of them the system is flooded. Here the fish remain for a day at 10 to 24° F. below zero and they are glazed with ice after freezing by dipping them in water. This ice retains the fish oil and keeps the flavor. After this operation they are wrapped in parchment paper and boxed for shipment at 10° F.

Oysters are held in cold storage at 35° F. for some time. After opening, the oysters may be placed in a bucket and frozen solid. This is not advisable.

Fruit. Fruits of all kinds are kept in cold storage, but the time in certain cases should not be long, since some of them lose flavor.

Apples are usually stored in $2\frac{1}{2}$ bu. barrels weighing about 150 lbs. and occupying 5 cu.ft. of space. They are usually held at 30 to 35° F. for winter apples while the softer summer apples are held about 5° above these. In England 29 to 30° F. has been used. Apples seem to improve on cold storage; there is a transformation of some of the starch into sugar.

Apples have been kept for months and even as long as two years. Care must be exercised in picking and packing. If carefully picked they keep for a considerable time. The shipment abroad is very extensive and the loss on cold storage apples is very slight, amounting to from about 25 cents to something over a dollar per barrel. This business in 1910 amounted to over nine hundred thousand barrels, valued at more than two million dollars.

Pears are improved by storage in a way similar to that for apples, but they are not usually kept so long. The temperature is about the same as that for apples or a little higher—30 to 36° F. These are usually placed in boxes of 40 lbs. weight when full and are picked in an under-ripe condition. Bushel crates are sometimes used. Closed barrels are not advisable, as the heat

cannot be removed from the center fast enough. Wrappers of paper are advisable to protect the fruit from bruises and to keep the color bright.

Peaches are kept for a few weeks only and are placed in boxes or crates weighing about 20 lbs. They lose flavor if held long. The storage is for the purpose of transportation and to lengthen the season for selling. They are held at from 32 to 36° F. In shipping and storing these the boxes are placed on top of each other about five boxes high, and the fruit should be cooled slowly and warmed slowly to prevent sweating. The fruit should be carefully picked and at times it is stored slightly under-ripe.

Strawberries lose their flavor on storage and they are kept for only a short time. They are held at 40° F. to prevent ripening. Experiments have been made which show that these and other berries may be kept for four weeks. Huckleberries have been held at 20° F. for pie making.

Plums may be kept for several weeks at 34° F. if firm and sound.

Grapes. These may be shipped from the West with success. They are held at various temperatures. Some require 32° F. while with others 34 to 36° is used. They should be dry when stored. They may be held from one to two months. Seventy days have been recorded for storage in redwood sawdust.

Oranges and lemons are important items in the commerce of California. This business amounted yearly in 1905 to over $25,000,000 or 30,000 car loads. These cars hold from 15,000 to 30,000 lbs. of fruit. The matter of storage has received close attention from the government as well as from private parties. Oranges are picked at convenient times from February to May and are usually shipped in crates 15×15×30 ins., weighing about 70 lbs. These are placed on end in storage and usually in two layers. The fruit must be carefully picked, as bruised fruit decays. They should be held in shipment at 32 to 50° F., and on account of the improper care the loss in shipment has amounted to over one million dollars per season. At 32° F. they may be stored three weeks to a month, but oranges are

uncertain for this time. At times the storage is extended to three months. They give off large amounts of gas and require ventilation and if the air is too dry shrinkage occurs.

Melons may be held for several weeks at 35° F. They must be carefully picked and selected for long storage. Usually the storage is for a short period.

Bananas are usually picked green and are allowed to ripen gradually, the amount of ventilation determining the speed of ripening. At 34° they may be stored for some time, while at 40° they gradually ripen. At 32° they are apt to turn black.

Vegetables. Potatoes are held at 35° F. in bags or barrels and should be so stacked that air may reach them. This must not be dry air. Potatoes are figured at 60 lbs. to the bushel. If in barrels there will be 5 cu.ft. to about $2\frac{1}{2}$ bushels. The room should be dark.

Tomatoes, if picked when just starting to redden, may be kept for two months. They are usually crated after wrapping in tissue paper. They are held at 40°.

Onions are stored at about 34° for six months. These give out an odor and should be kept in a special room. They are placed in bags or barrels.

Celery is held in crates of about 140 lbs. These crates are about 24×24×30 ins. Celery is held at about 34° F. for three or four months. The seasons for production in different parts of the United States make it possible to get this at all times of the year.

Cabbages are held at 35° F. These are stored in barrels or crates. Air circulation is necessary.

Tobacco and cigars are held at 40 to 45° F. and will retain their flavor if kept in one condition. This low temperature prevents the development of insect life.

Furs, Rugs and Clothing. The matter of the cold storage of goods subject to moths and other insects received attention during the last decade of the nineteenth century. In a paper by Dr. L. O. Howard it was stated that insects caused a loss in cereals of one hundred million dollars per year, and Mr. A. M. Reed conceived the idea of preventing a similar loss from insects

acting on furs and woolen goods and experimented on the eggs of the moth and buffalo beetle and found that 50 or 55° F. was sufficient to prevent the hatching and 40° prevented the

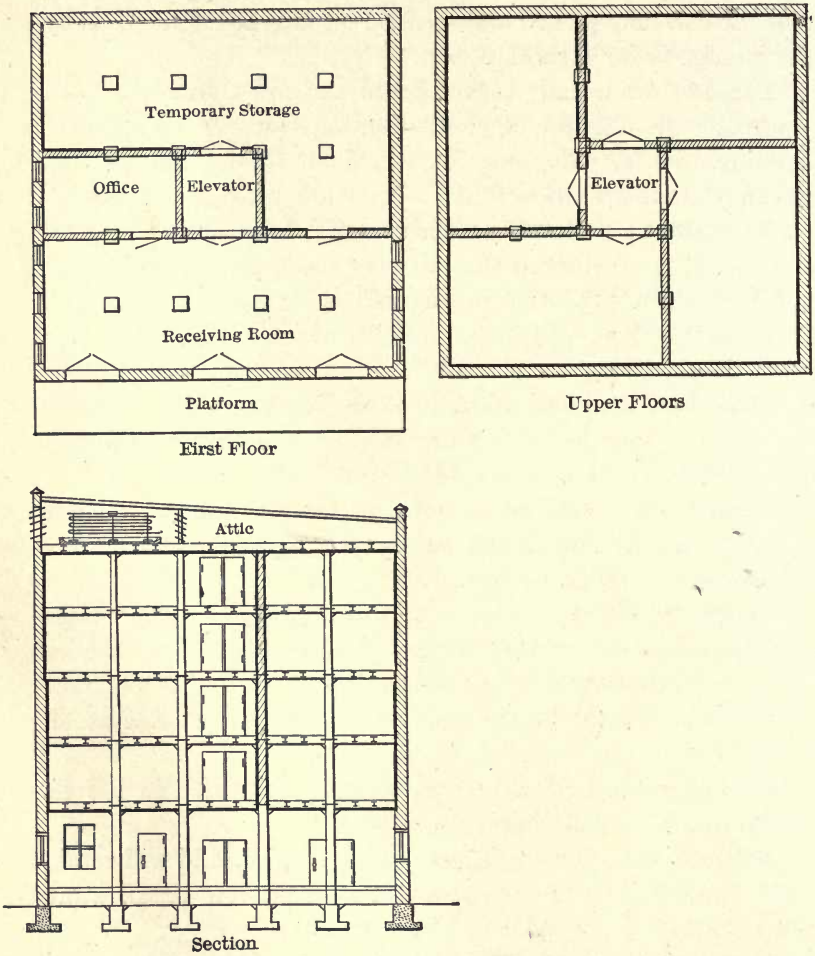


FIG. 116.—Small Store House.

passing from the larval state. The miller and the beetle were killed at 32° F. and in the center of rugs they were killed in several weeks at temperatures of 32 to 40° F. This led to the establishment of cold storage for such articles held at 32°.

Florists hold lily of the valley pips, lily bulbs, ferns, smilax and other plants or bulbs for months at low temperature. They also use the **cold-storage room to control the growth** of plants.

The **construction of cold-storage warehouses** will vary with the peculiarities of the designer and the requirements of the ground selected for the plant. In general there is a receiving room near a railroad track and a truck platform as well, close to the office for the receipt and delivery of goods. In some cases there is only a railroad platform, as goods are handled

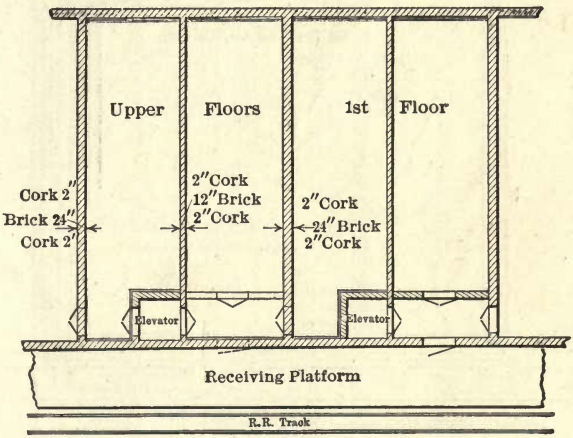


FIG. 117.—Plan of Large Store House.

each way in carload lots. The receiving room is sufficiently large to hold several transhipping hand trucks and is connected by elevators to the various floors. In **small houses** the elevator shaft could pass through the center of the house opening into four rooms placed around it on each floor. The separate rooms are required to give the necessary number of rooms for the differing foodstuffs to be stored. This plan is shown in Fig. 116. In this way the elevator serves the four rooms. The attic under the roof is used for the condensers and storage and also serves as a heat insulator. The cork insulation on outside walls, certain inner walls, ceiling of first floor and on the upper ceiling is shown by heavy lines.

For **extensive plants**, as those built in cities, the rooms may be larger and extend over the complete space of one floor.

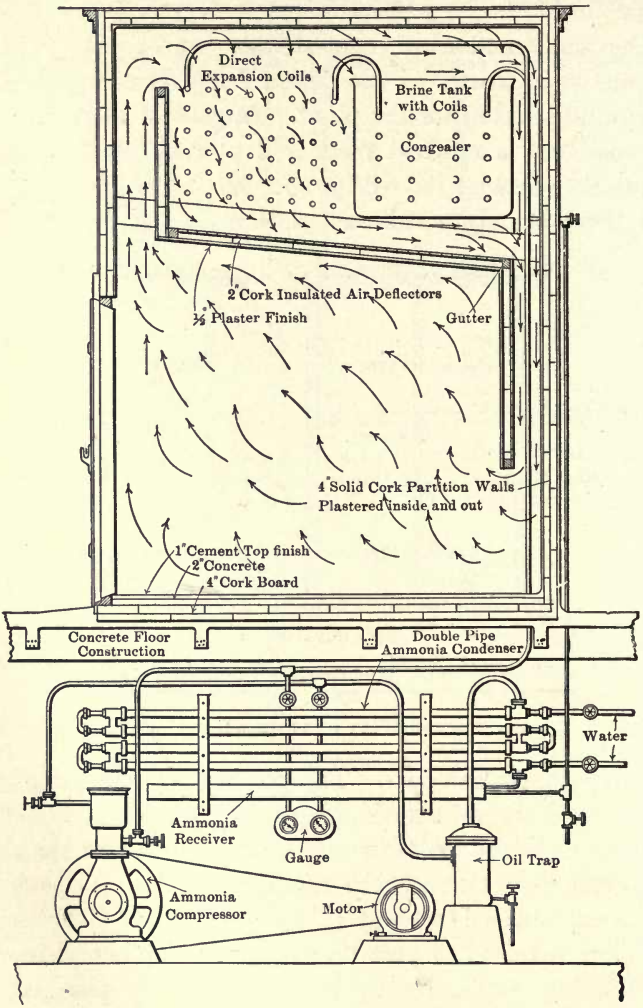


FIG. 118.—Arctic Cooling Plant for Store.

Such a plant is shown in Fig. 117. The elevators in this plan serve two houses or two rooms of one house. A **small storage room for a market** is shown in Fig. 118 in which the various

details may be seen easily. The use of a **brine tank** or **congealer** makes it possible to shut down the machine at night after freezing the brine, this frozen brine furnishing the refrigeration during that time. The various parts of the plant

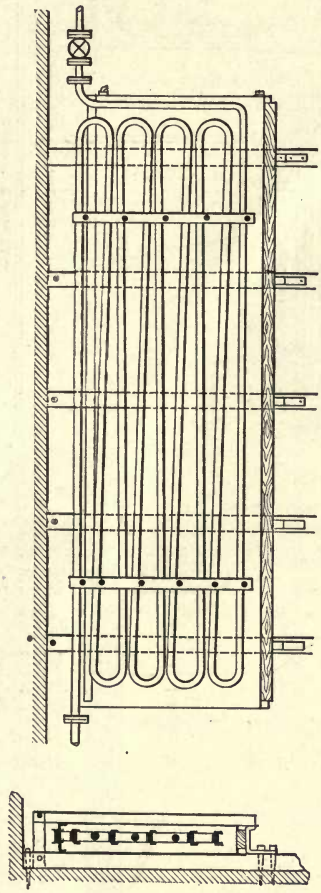


FIG. 119.—De la Vergne Congealer.

may be traced out. The **congealer**, Fig. 119, for wall coils, is sometimes used in larger rooms.

Figs. 120, 121 and 122 show the arrangement of pipes, insulation room and machinery for small plants. In Figs. 118 and 122 the air circulation may be followed.

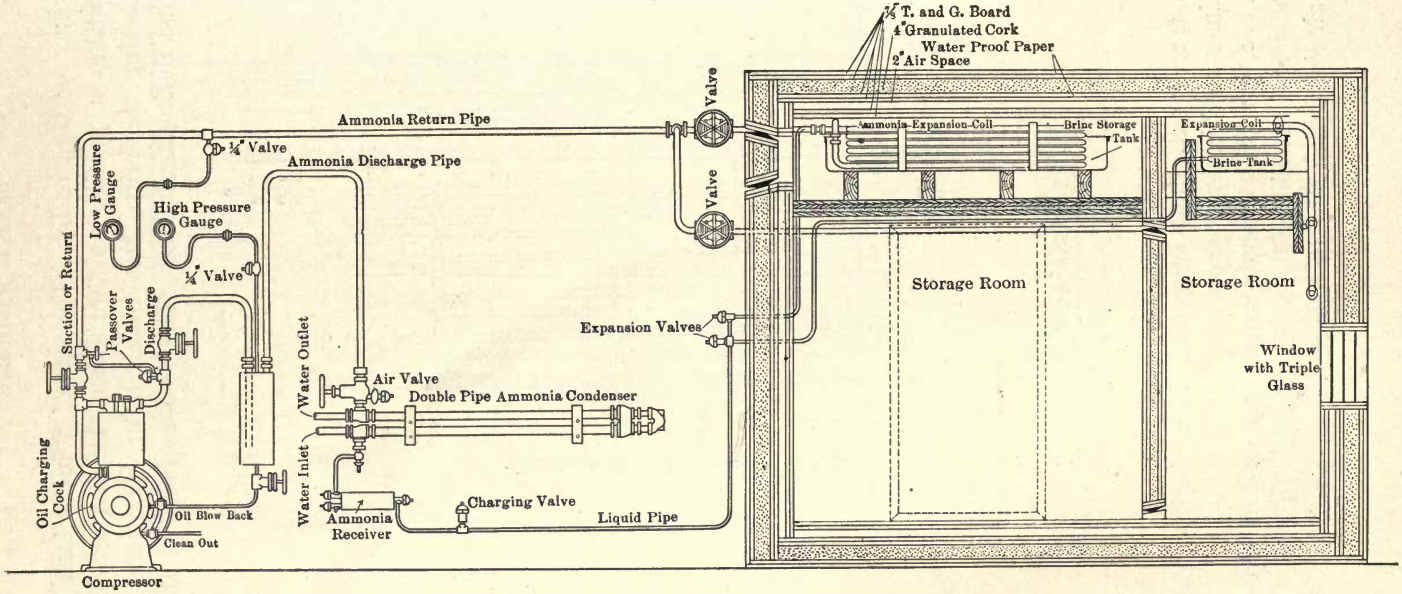


FIG. 120.—Remington Direct-expansion Refrigerating Plant.

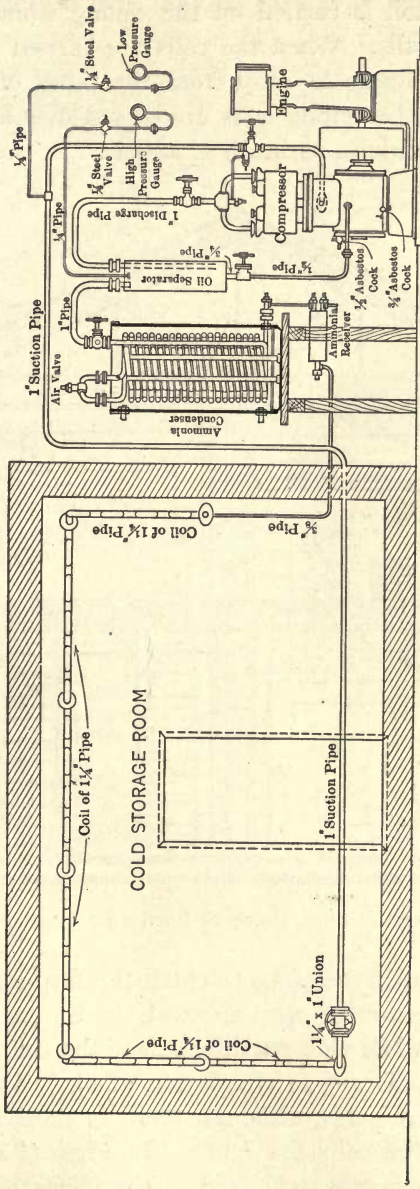


FIG. 121.—Remington Small Refrigerating Plant.

The various **arrangements of piping** are shown in Fig. 123. In Fig. *A*, the coil is carried on the ceiling while in *B* it is carried on the walls. When the coils are carried on the ceiling, moisture is likely to drop from the pipes on the goods below, and then the ceiling coils are placed over aisles or else they are placed in lofts as in *D*, *E* and *F*. In these lofts the

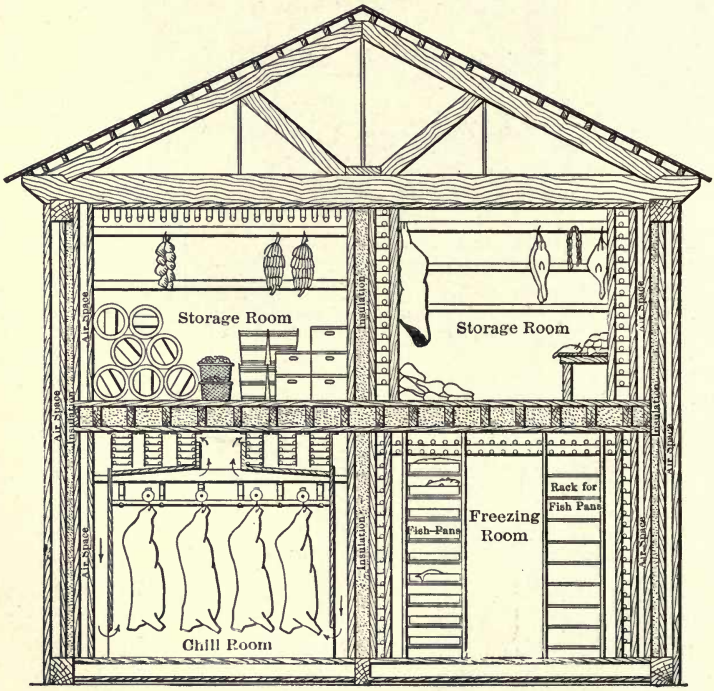


FIG. 122.—Small Store House of Remington Machine Co.

floor is placed under the pipes to catch the drip and take it to a gutter, but there is an open space at the center to allow the cold air to fall, while side partitions near the center or at the sides near the walls aid in circulation of air. The wall coils are better in most cases, although there is danger of cooling the goods near the coils too much. In Figs. *C* and *E* there are brine tanks or congealers used in the rooms and the cooling of this large amount of brine to a low temperature or the

freezing of the brine permits the compressor to be shut down. Figs. *A*, *B* and *D* may be used with brine or direct expansion

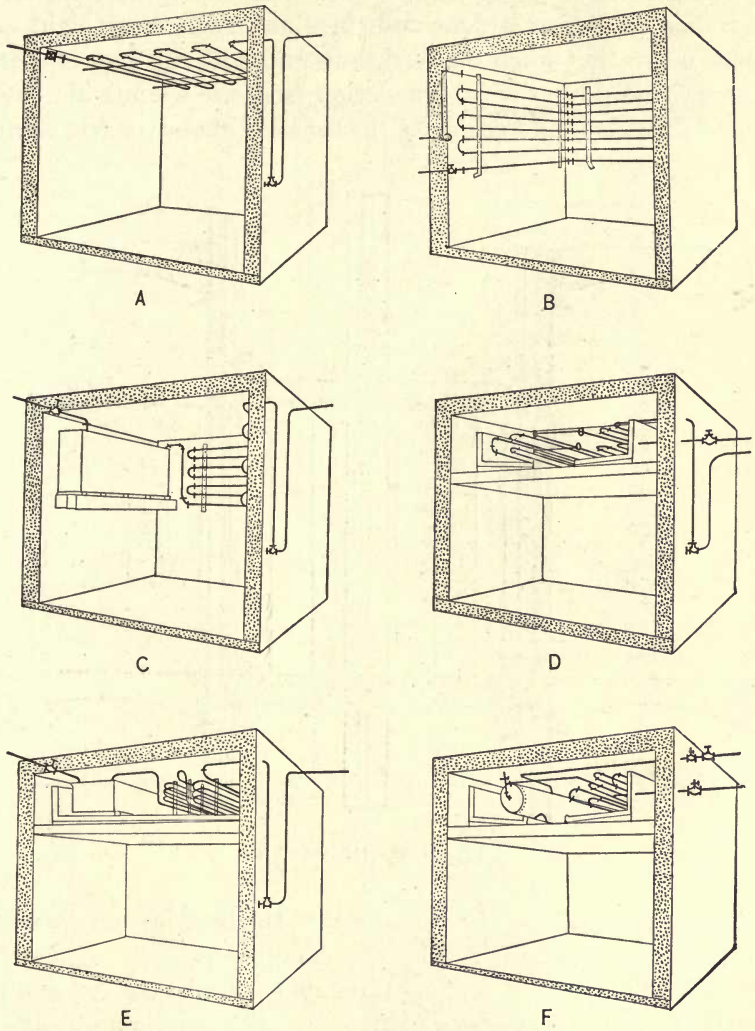


FIG. 123.—Arrangement of Piping as Shown by Creamery Package Co.

of ammonia. *C* and *E* are for direct expansion. For intermittent operation when brine is to be circulated, Fig. *F* is employed, the tank supplying the refrigeration on shutting

down the plant. Such an arrangement is used when ammonia piping is objectionable.

In constructing the **elevator** and **well** the elevator car has a ceiling as well as a floor and these are made as air tight as possible, so that when the refrigerator door at any floor is open there is no danger of air circulation from the warmer air outside. The floor and ceiling of the car have rubber or felt filling

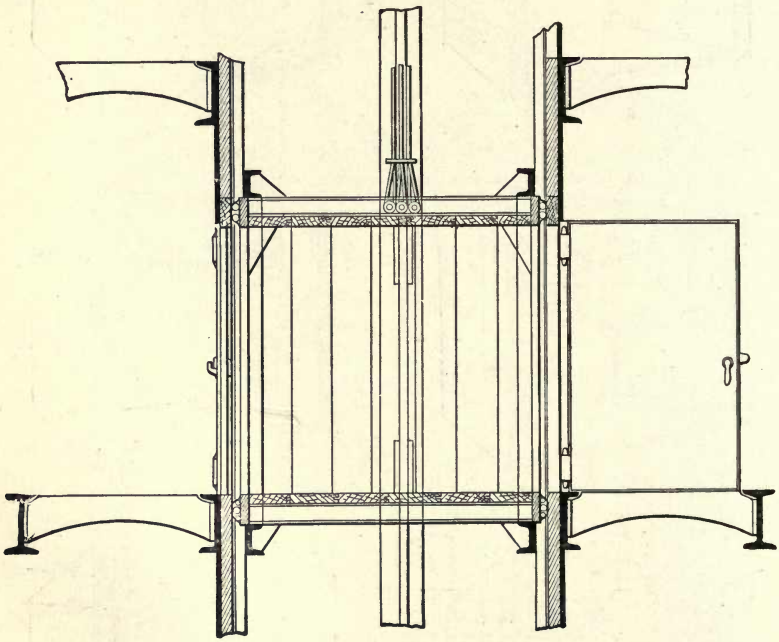


FIG. 124.—Elevator.

strips, closing off the air space so that the heat loss on opening the door is a minimum. Fig. 124 shows this.

Fig. 125 gives a section through a **storehouse for meats** while Fig. 126 illustrates a section of a ship containing refrigerated space. The arrangement of the cooling coils for circulation and drip is to be noted as well as the air loft under the roof as in Fig. 116.

The number of **rooms used in hotels** varies with the hotel.

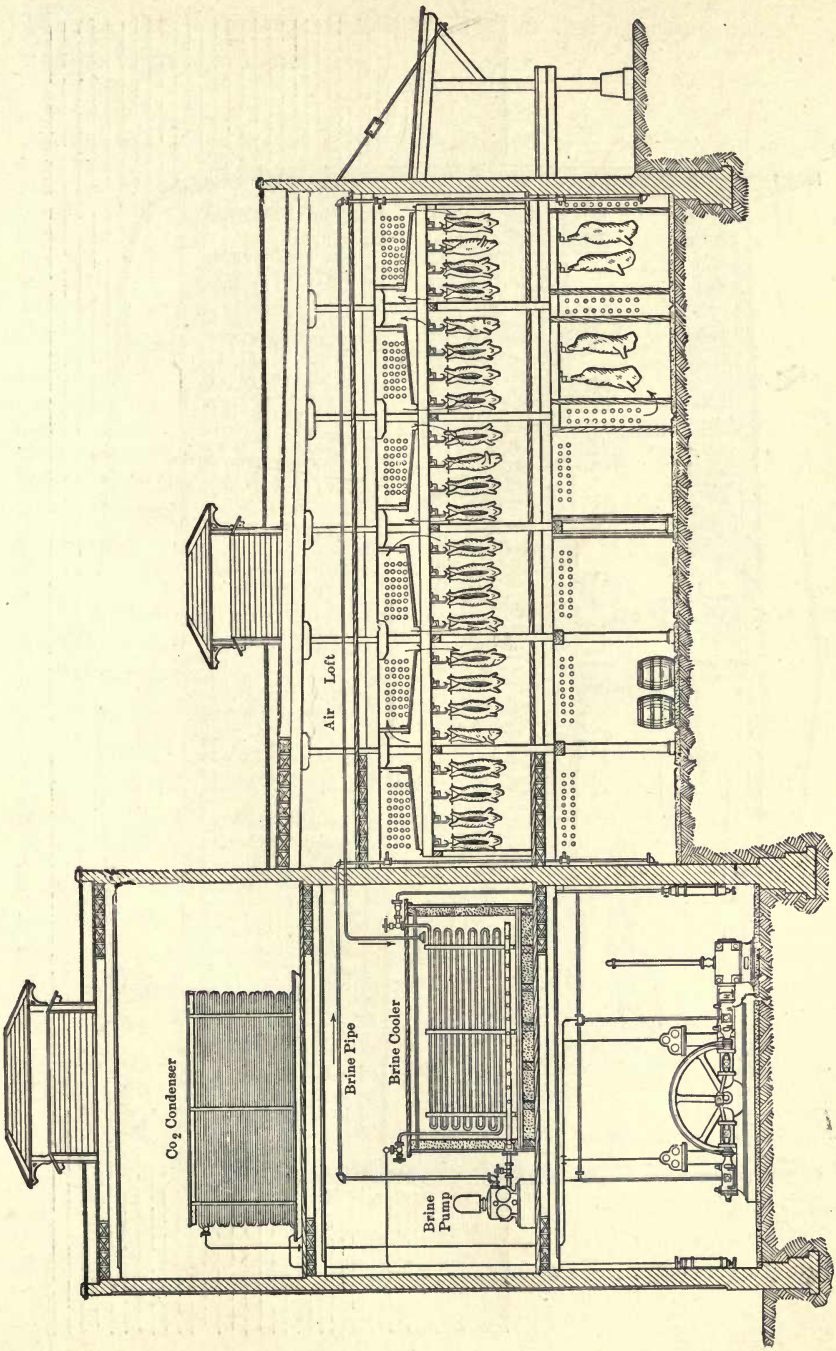


FIG. 125.—Packing House with Brine System. Kroeschell & Co.

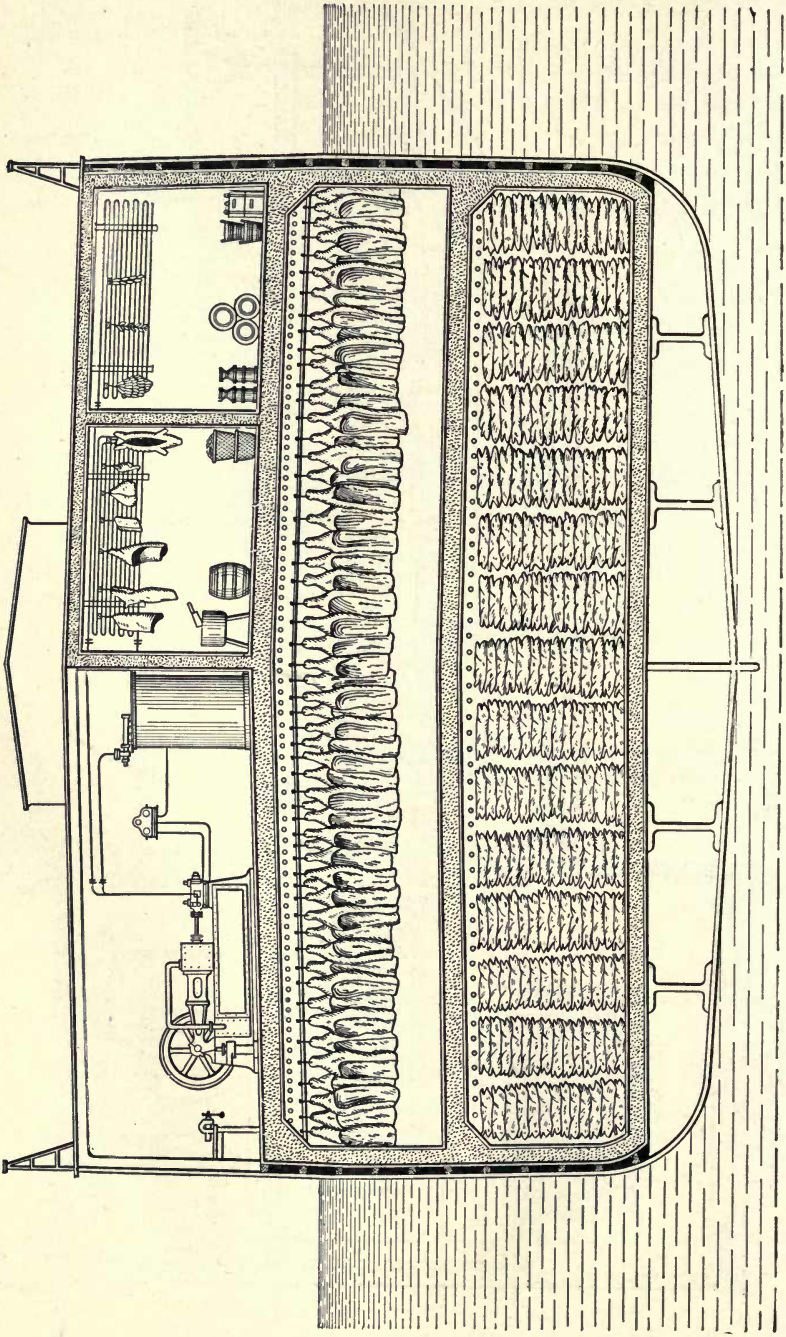


FIG. 126.—Ship with CO_2 Machine of Kroeschell & Co.

Thus in the Blackstone Hotel of Chicago the following cold-storage rooms are found:

Basement.

Vegetable box.....11′ 5″×10′ 3″×7′
Fruit box.....13′ 9″×10′ 3″×7′
Meat box.....14′ 4″×8′ 4″×7′
Bouillon box.....7′ 2″×8′ 4″×7′
Game box.....5′ 6″×8′ 4″×7′
Egg box.....4′ 6″×4′ 2″×7′
Butter box.....4′ 6″×3′ 2″×7′
Milk box.....9′×5′ 2″×7′
Cheese box.....5′ 4″×3′ 3″×7′
Oyster box.....12′ 0″×6′ 8″×7′
Fish box.....6′×6′ 8″×7′
Fine wines box.....7′ 9″×4′ 7″×7′
Ice-cream box.....10′ 0″×10′ 0″×10′
Sharp freezer box, 5′ 0″×10′ 0″×10′ 0″
Draft beer box.....9′ 0″×9′ 0″×6′

Banquet Hall.

Refrigerator.....14′×4′×9′

Kitchen.

Poultry box.....5′×3′×3′ 10′
Fish box.....9′×3′×3′ 10′
Lobster box.....5′×3′×3′ 10″
Bouillon box.....2′ 8″×2′ 8″×2′ 10″
Cold plate box.....13′×3′ 4″×3′ 10″
Cold plate box.....11′ 3″×4′×3′ 10″
Cold meat box.....4′×4′×9′
Sandwich box.....6′×3′×9′
General kitchen.....9′×13′×9′
Oyster box.....16′×4′×9′
Pantry.....6′×6′×9′
Freezers for ices.....6′×3′×7′
Fruit and salad box.....5′×3′×9′
Milk box.....4′×3′×3′
Baker's box.....8′×8′×8′
Ice cream.....11′ 6″×3′ 6″×3′
Short order.....5′×3′ 4″×3′ 10″
Cook's box.....16′×4′×9′

In addition to the above there are some boxes on the dining-room floor and the club floor.

To do this work and to cool the dining rooms and certain places as well as to make some ice requires a 50-ton machine and a 75-ton machine.

In Fig. 127 a direct expansion CO_2 **plant for a brewery** is shown. Each storehouse is cooled by large direct-expansion coils. In the air loft above the **fermenting tanks** is noted a **sweet-water cooler.** The water which is cooled in this tank is passed through coils in the fermenting vats to remove the heat of fermentation and control this process. The **large vats** are used for proper aging before storage in the **chip casks** on the lower floor. The path of the gas from the two compressors through the condenser and piping and the construction of the walls should be examined.

The **refrigeration** for **storage of food** may be done with natural ice and salt as was mentioned in Chapter II. In Minnesota a wholesale and retail market refrigerates 40,000 cu.ft. of space with 553 tons of ice at $1.65 per ton and 67 tons of salt at $7.00 per ton, holding the room at 15° F.

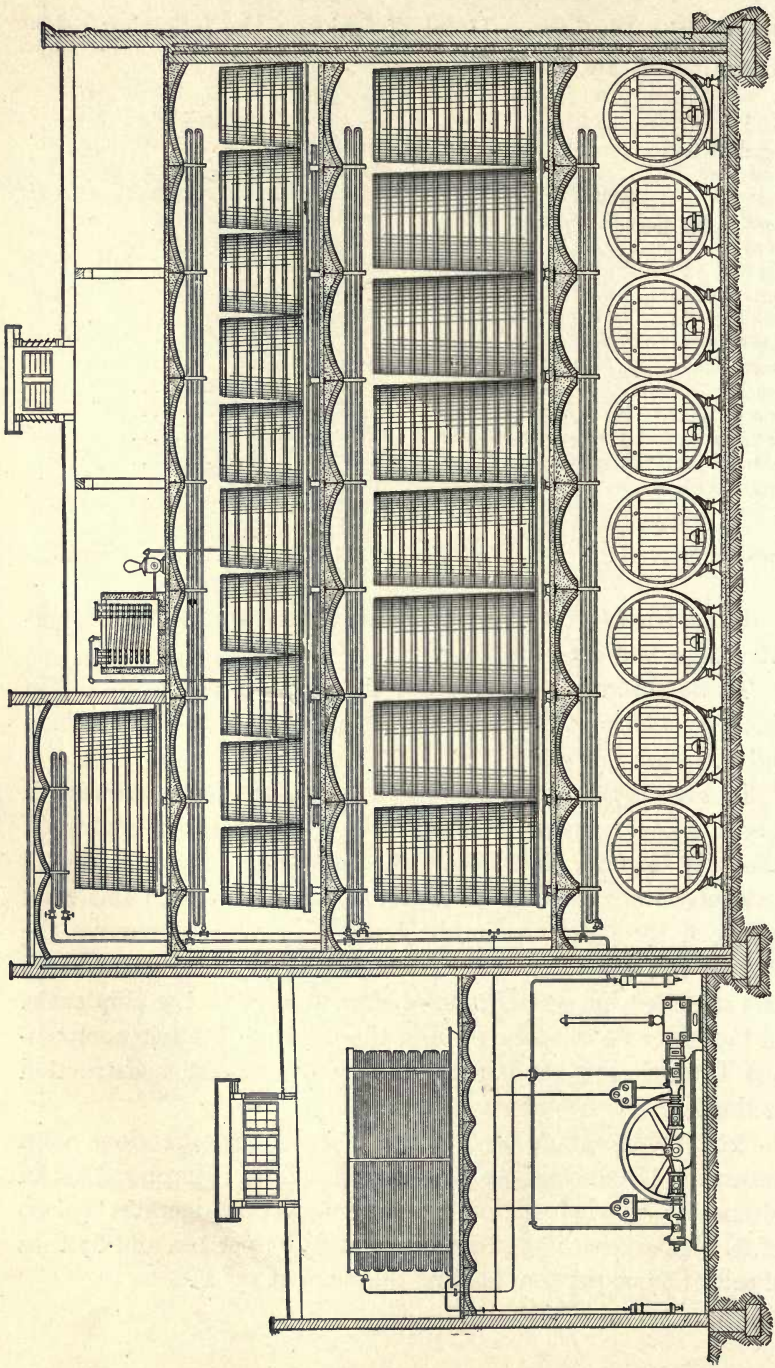


FIG. 127.—Brewery Equipped with Kroeschell Brothers CO_2 Apparatus.

The **walls of these houses and the floors** are constructed in various ways. Thus at the **Boston Terminal Refrigerating Co. plant,** a building 156 by 100 ft. with seven floors and a ventilating loft, the walls were made of 4-in. vitrified waterproof brick, 8 ins. of hollow tile and two 2-in. thicknesses of Nonpareil cork held in place by asphalt cement. This wall does not carry any of the floor load. The building is of reinforced concrete and the only floors insulated for heat are the basement floor, third floor, and eighth floor. These floors are insulated with two thicknesses of 2 in. or 2- and 3-in. thicknesses of cork board with an asphalt wearing surface. This construction forms a fireproof building. A suggestion has been made to use two 4-in. tiles with a third tile 8-ins. from the outer ones with a cork filling between the outer wall for a fireproof building. This wall carries no weight. The columns carrying the weight are of reinforced concrete. In this system the partitions are made of two 4-in. tiles with a 6- or 8-in. cork fill. The floors are of 6 ins. reinforced concrete, 2 or 4 ins. of cork and asphalt on top. The doors are covered with iron.

In an apple-storage warehouse the walls were made of 2×6 hemlock boards laid flat, as are used in grain elevators, and faced on one side with 4 ins. of brick. Fig. 128. Such a construction should give good results. The installation cares for 15 cu.ft. of space with 1 lin.ft. of 2 ins. direct expansion.

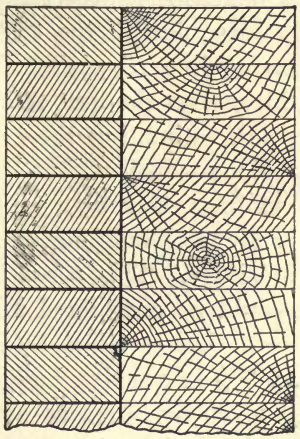


FIG. 128.—Grain Bin Construction for Apple Storage.

In **large plants** the insulation is such that the **heat loss is 1.8 B.t.u. to 0.6 B.t.u. per square foot per hour** or for values of K of 0.03 to 0.01.

Any of the forms of Chapter VI may be used for the insulation and for any form of construction the value of K may be computed as shown.

Partitions between rooms may be made of two 2-in. cork boards faced with 4-in. tile or with cement plaster.

The **temperature of the rooms should receive considerable thought** in determining what should be used. The highest possible temperature for good storage should be used. Some storage men claim that zero rooms cost from 50 to 75% more to operate than rooms at 30° F. The proper selection may make a success from what has been a failure. The table on p. 215 gives the temperatures required for different articles.

After the temperature is fixed, the **amount of insulation should be figured so as to make the annual expense a minimum.** The annual expense is made up of interest, depreciation, taxes, insurance on insulation, value of space occupied and insurance on the stored materials and the cost of absorbing the heat leaking through the insulation. If the thickness is increased and the kind of insulation improved, the first items will increase, but the cost of absorption will be decreased and if the sum of these is decreased then the improvement pays. If the sum is increased a poorer insulation would give better results, the increase due to the cost of insulation not making up for the saving in refrigeration. By plotting these costs for different thicknesses the best thickness may be found.

In estimating these items the charge in insurance due to various constructions must be considered. Thus in frame buildings, according to J. H. Stone, the insurance is 1% while in fireproof buildings it is only $\frac{1}{4}$% and $\frac{1}{2}$% for semi-fireproof buildings. This refers to goods as well as buildings and this should be considered in fixing the cost. The depreciation on insulation is taken by him at 4% in good construction and at 8% in wooden buildings. The cost of insulating material is given by Stone as 27 cents per square foot for 2 B.t.u. per square foot per twenty-four hours per degree difference.

The **cooling is accomplished by coils or by air circulation.** The coils, as shown in Fig. 123, are either brine coils or direct-expansion coils. In some plants brine is thought to be necessary. In present-day work especially with welded pipes ammonia is safe and is used.

The **direct-expansion system** requires less difference in pressure between suction and discharge main, giving a more efficient plant. The **brine system,** however, in addition to the questionable advantage of safety in case of rupture does possess certain

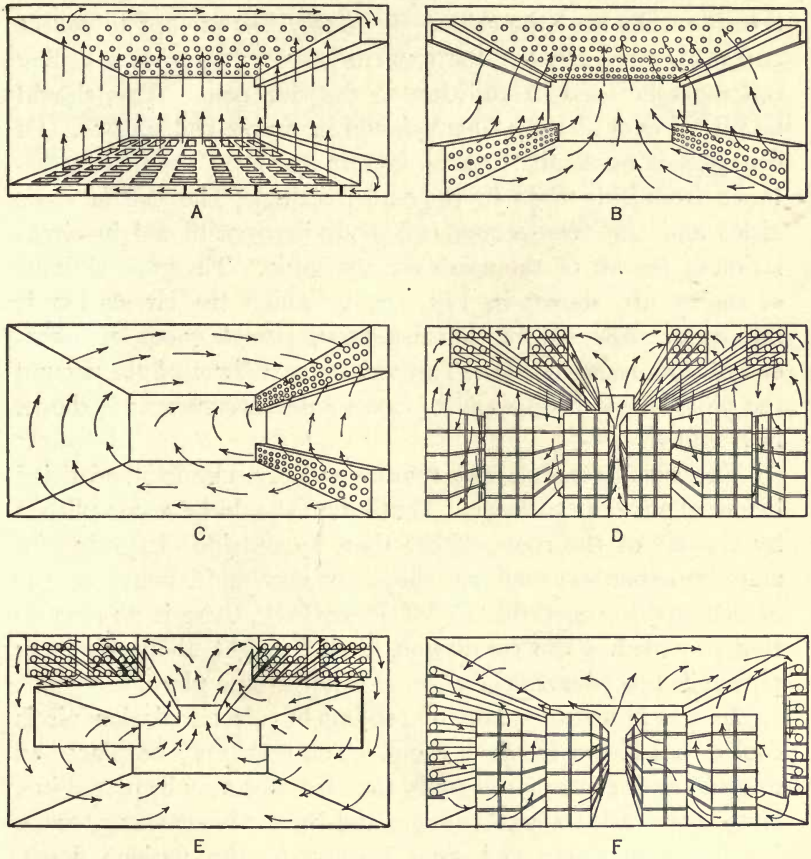


FIG. 129.—Arrangements of Direct and Indirect Refrigeration.

advantages. If a large amount of brine is cooled during the day this may be used when the compressor is shut down and, moreover, the cost of the expensive ammonia to fill the system is eliminated. To increase the storage capacity it is even possible to freeze the brine, removing about 150 B.t.u. per pound

of brine. These advantages are worthy of consideration, although the lower back pressure on the compressor when brine is used to care for the double transfer of heat makes the advisability of its use a matter hard to determine.

The coils, of whatever kind they may be, are best placed on the side walls near the ceiling, as ceiling coils are apt to collect and drop moisture. If the room is over 25 ft. wide a ceiling coil must be used in addition to the side coils. This should be placed over aisles. There should be ample coil surface. Of course it is necessary to keep certain goods at the proper distance from side coils to prevent freezing. The use of cross aisles and the arrangement of goods in tiers to aid in circulation of the air of the room are advisable. The cross-sections of rooms are shown in Fig. 129 in which the circulation is indicated. The use of the aisles to separate goods of different owners is advisable. The use of partitions of tin around the pipes to catch drip and to cause a definite current is shown in *E* and *F*.

The **air for the storage room** should be cleaned and dried before allowing it to enter. Ventilation should be accomplished by the air of the room rather than by outside dirty air. In many storehouses such as the 4,300,000 cu.ft. house of the Merchants' Refrigerating Co. of Jersey City, there is no circulation provided. The circulation is all brought about by large pipe coils placed over the aisles and not at one place.

In the indirect system of cooling by air circulation air is cooled and blown into the room. The coils may be placed in ducts in ceiling or as is generally the case they may be placed in a large space called a bunker room and the air blown across these is carried to ducts and flues leading to the various floors. The **distribution of air** in the storage room is difficult. In some cases it is distributed through numerous small holes in the ceiling and the warmed air is removed through holes in the floor. In this way an even distribution over the whole room is obtained, although other methods are used. The air may be recirculated if there are no odors. This is one of the objections to the indirect system. Smoke or odors from

one room may contaminate the stored goods in another. Fig. 129 illustrates warehouse rooms using forced-air systems.

The **bunker room** is shown in Fig. 130. In this, brine or a volatile liquid is passed through the coils and abstracts heat from the air, which is blown across the pipes. The moisture

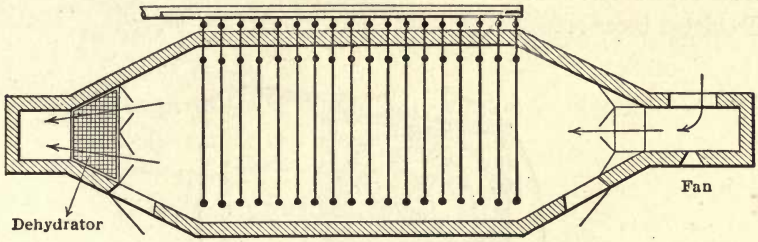


FIG. 130.—Bunker Room.

removed from the air freezes on the outside of the pipes, but by circulating brine occasionally over the pipes this frost is removed or warm brine may be turned in when the air is shut off.

In some houses the radiation surface is increased by the

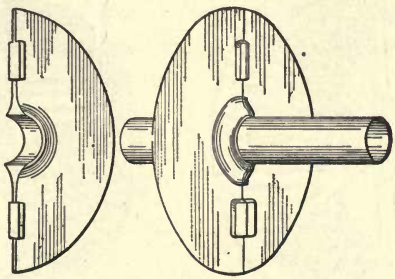


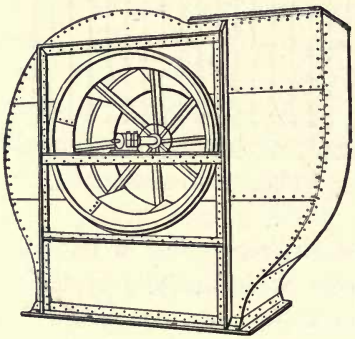
FIG. 131.—Radiation Discs.

use of split discs added to the outside of direct-expansion pipes, Fig. 131. These are not used often at present although in former times they were used extensively.

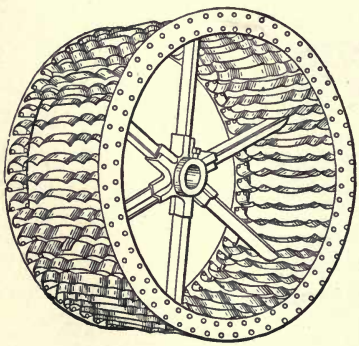
The air is driven by a fan blower. Forms of blowers are shown in Fig. 132. The **size of the fan and the power** to drive the fan are fixed after the sizes of ducts

have been computed. The **duct sizes** are fixed by the allowable velocities of the air. **The velocities** to be used are as follows:

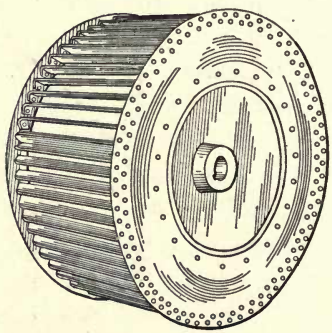
Main ducts	1200 ft. per min.	= 20 ft. per sec.	
Branch ducts	800 "	= 13 "	
Register faces	300 "	= 5 "	



Buffalo Planoidal Fan.



Sturtevant Multivane.



American Sirocco.

FIG. 132.—Fan Blowers.

The **quantity of air** is fixed by finding the quantity of heat to be removed and the allowable rise in temperature in the air. If Q is the heat entering the room per hour, t_r is the temperature of the room and t_0 is a temperature to which the

air can be cooled in the bunker room, the volume of air per hour V is given by

$$V = Mv = \frac{Qv}{0.24(t_r - t_i)} = \frac{Q}{0.02(t_r - t_i)}. \quad \ldots \quad (1)$$

V = cubic feet of air per hour;
Q = heat removed per hour in B.t.u. figured from methods of Chapter V;
t_i = temperature at outlet from bunker in deg. F.;
t_r = temperature of room in deg. F.;
v = volume of 1 lb. of air
= 12 cu.ft. approximately.

If w equals the velocity in feet per second in the duct or register the area F in square feet is given by

$$F = \frac{V}{3600w}. \quad \ldots \quad (2)$$

The sizes of the ducts are found and then the **hydraulic radius** of each duct is computed. This is equal to the cross-section of the duct divided by the perimeter. If b and d are the dimensions of the duct in feet and R_1 is the hydraulic radius in feet the following are true:

$$bd = F, \quad \ldots \quad (3)$$

$$\frac{bd}{2(b+d)} = R_1. \quad \ldots \quad (4)$$

The **friction loss** in pressure along a straight pipe of L ft. length is given in feet of air pressure by

$$H = 0.02 \times \frac{L}{4R_1} \frac{w^2}{2g}. \quad \ldots \quad (5)$$

For bends of radius equal to $2b$, the loss is

$$H = 0.15 \frac{w^2}{2g}. \quad \ldots \quad (6)$$

In grills the loss is given by

$$H = 0.8\frac{w^2}{2g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

At gradual changes of section there is no loss.

For loss in the bunkers with staggered cooling coils the loss of head may be taken as

$$H = 0.4\frac{w^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

for each coil or line of pipe over which the air must travel.

If now in any line the sum of the various losses is taken, the total loss may be found. This is the **static pressure** to be produced by the fan and in some fans this may be taken as about three times the velocity pressure, $\frac{w^2}{2g}$. The sum of these two pressures is called the dynamic pressure, and the ratio of this to the static pressure is usually given in catalogs of fan makers. The static pressure varies from 70 to 90% of the dynamic pressure in various forms. The dynamic pressure is the one listed in most catalogs. Having the dynamic pressure in feet of air found by dividing the static pressure by 0.75, this may be changed to ounces per square inch by dividing by 120 for air at 70° or by 110 for air at 20° F. If pressure in inches of water is desired this pressure in ounces per square inch is multiplied by 1.73.

The data for three types of fan are given in the table on p. 251 for the points of best efficiency:

From this table the fan must be picked out to deliver the desired volume, at a given dynamic pressure.

Since in general the pressure is not a tabular pressure, the equivalent volume must be found for a tabular pressure. When a fan is run at the same point of its efficiency curve, the pressure varies as the square of the velocity or as the square of the number of revolutions; the quantity of discharge varies as the velocity or number of revolutions and the power which

Fan Data

Sirocco Fan. Velocity Press. = 0.288 Dyn. Press.					Buffalo Conoidal Fan. Velocity Press. = 0.225 Dyn. Press.				Sturtevant Multivane Fan. Velocity Press. = 0.10 Dyn. Press.			
Size.	Diam. of Fan Wheel.		Dynamic Press.		Size.		Dynamic Press.		Size.		Dynamic Press.	
			½ oz.	1 oz.			½ oz.	2 oz.			½ oz.	1 oz.
00	3	Cu.ft.	38	77	30	Cu. ft.	1,720	3,435	3	Cu.ft.	1,000	2,000
		R.P.M.	2,290	4,580		R.P.M.	632	1,255		R.P.M.	502	1,003
		B.H.P.	0.005	0.037		B.H.P.	0.39	3.11		B.H.P.	0.12	1.0
0	4½	Cu.ft.	87	175	35	Cu.ft.	2,340	4,675	4	Cu.ft.	1,448	2,895
		R.P.M.	15,24	4,580		R.P.M.	545	1,085		R.P.M.	418	835
		B.H.P.	0.011	0.084		B.H.P.	0.53	4.23		B.H.P.	0.19	1.5
1	6	Cu.ft.	155	310	40	Cu.ft.	3,060	6,100	5	Cu.ft.	1,950	3,900
		R.P.M.	1,145	2,290		R.P.M.	482	960		R.P.M.	359	717
		B.H.P.	0.018	0.147		B.H.P.	0.69	5.52		B.H.P.	0.25	2.0
1½	9	Cu.ft.	350	700	45	Cu.ft.	3,890	7,760	6	Cu.ft.	2,565	5,130
		R.P.M.	762	1,524		R.P.M.	422	845		R.P.M.	314	627
		B.H.P.	0.042	0.333		B.H.P.	0.88	7.03		B.H.P.	0.33	2.6
2	12	Cu.ft.	625	1,250	50	Cu.ft.	4,780	9,560	7	Cu.ft.	4,000	8,000
		R.P.M.	572	1,145		R.P.M.	378	753		R.P.M.	251	501
		B.H.P.	0.074	0.588		B.H.P.	1.09	8.67		B.H.P.	0.5	4.0
3	18	Cu.ft.	1,410	2,820	60	Cu.ft.	6,875	13,650	8	Cu.ft.	5,800	11,600
		R.P.M.	381	762		R.P.M.	318	636		R.P.M.	218	435
		B.H.P.	0.167	1.33		B.H.P.	1.55	12.38		B.H.P.	0.72	5.8
4	24	Cu.ft.	2,500	5,000	70	Cu.ft	9,450	18,750	9	Cu.ft.	7,850	15,700
		R.P.M.	286	572		R.P.M.	272	542		R.P.M.	180	359
		B.H.P.	0.296	2.35		B.H.P.	2.14	17.00		B.H.P.	0.99	7.9
5	30	Cu.ft.	3,910	7,820	90	Cu.ft.	15,600	31,200	10	Cu.ft.	10,300	20,600
		R.P.M.	228	456		R.P.M.	210	419		R.P.M.	157	314
		B.H.P.	0.460	3.68		B.H.P.	3.54	28.3		B.H.P.	1.25	10.0
6	36	Cu.ft.	5,650	11,300	110	Cu.ft.	23,100	45,700	11	Cu.ft.	12,925	25,850
		R.P.M.	190	381		R.P.M.	172	343		R.P.M.	139	278
		B.H.P.	0.655	5.30		B.H.P.	5.24	41.4		B.H.P.	1.64	13.1
8	48	Cu.ft.	10,000	20,000	130	Cu.ft.	32,400	64,700	12	Cu.ft.	16,000	32,000
		R.P.M.	143	286		R.P.M.	146	291		R.P.M.	126	251
		B.H.P.	1.18	9.40		B.H.P.	7.34	58.6		B.H.P.	2.02	16.2
10	60	Cu.ft.	15,650	31,300	150	Cu.ft.	43,000	86,000	13	Cu.ft.	19,400	38,800
		R.P.M.	114	228		R.P.M.	127	253		R.P.M.	114	2.28
		B.H.P.	1.84	14.7		B.H.P.	9.80	78.0		B.H.P.	2.44	19.5
12	72	Cu.ft.	22,600	45,200	170	Cu.ft.	55,500	110,000	14	Cu.ft.	23,200	46,400
		R.P.M.	95	190		R.P.M.	112	235		R.P.M.	105	209
		B.H.P.	2.66	21.2		B.H.P.	12.55	88.8		B.H.P.	2.94	23.5
14	87	Cu.ft.	30,800	61,600	190	Cu.ft.	69,000	137,800	15	Cu.ft.	27,200	54,400
		R.P.M.	81	163		R.P.M.	100	199		R.P.M.	97	194
		B.H.P.	3.61	28.9		B.H.P.	15.61	124.8		B.H.P.	3.44	27.50
15	90	Cu.ft.	35,250	70,500	200	Cu.ft.	76,600	152,500	17	Cu.ft.	36,150	72,300
		R.P.M.	76	152		R.P.M.	94	188		R.P.M.	84	168
		B.H.P.	4.14	33.1		B.H.P.	17.35	138.0		B.H.P.	4.42	35.4

varies as the product of quantity and pressure will vary as the cube of the number of revolutions. Hence

$$V_e = V_a \frac{N_e}{N_a} = V_a \sqrt{\frac{P_t}{P_a}}. \quad \ldots\ldots (9)$$

V_e = equivalent volume discharged at speed N_e revolutions per minute;
V_a = actual volume discharged at speed N_a;
N_e = revolutions per minute to give total dynamic pressure P_t;
N_a = revolutions to give pressure P_a;
P_t = tabular pressure;
P_a = actual pressure.

Having V_e the fan may be selected and then N_a may be found to give the proper pressure and quantity.

$$N_a = N_e \sqrt{\frac{P_a}{P_t}}. \quad \ldots\ldots\ldots\ldots (10)$$

The power to drive this fan is given by

$$HP_a = HP_t \frac{V_a P_a}{V_e P_t}; \quad \ldots\ldots\ldots\ldots (11)$$

HP_a = actual horse-power to drive fan;
HP_t = tabular horse-power to drive fan.

If the relation between static pressure and velocity pressure is changed from that used, these values are changed, and although the tables may be used to get equivalent quantities, there is little use in giving the method of doing this, as the fan would then be working inefficiently.

The fan and its power to be used are now known and the dimensions may be found in the tables on pp. 253 and 254.

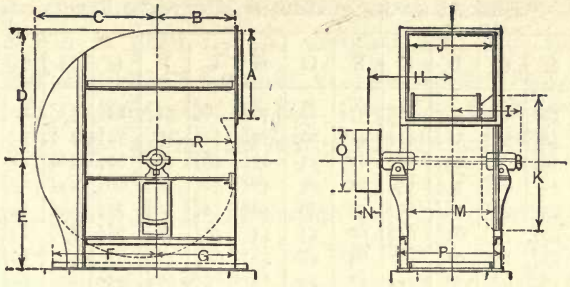


FIG. A.—Fan Dimensions.

DIMENSIONS OF SIROCCO FAN IN INCHES

Size.	A	B	C	D	E	F	G	H	I	J	K	N	O	P
7	28	25¼	40	42	36	31½	27½	26½	23½	28	44	8	20	31½
8	32	28¾	45⅞	48	40½	36	31	29	25½	32	50	9	22	35½
9	36	32⅜	51⅜	54	45½	40	34½	34	30¾	36	56	9	24	40
10	40	36	57¼	60	50½	44½	38	37	33½	40	62	10	26	44
11	44	39½	62⅝	66	55	48	41½	39½	36	44	68	10	28	48
12	48	43	68¼	72	59½	52½	45	45	41¼	48	74	11	30	52
13	52	46½	74½	78	64½	57½	49½	48	44¼	52	80	11	32	56
14	56	50½	80	84	69	59½	51	52	47¾	56	86	12	34	56
15	60	54	85⅝	90	73½	64½	54½	53	49¼	60	92	12	36	60

DIMENSIONS OF BUFFALO CONOIDAL FAN IN INCHES

Size.	A	B	C	D	E	F	G	H	I	K	N	O	P
30	11	10½	14¾	16⅞	13⅝	14	12½	12	9½	17¼	3	7	15 3/16
35	12⅞	12¼	17¼	19¾	15¾	16	14¼	13	10½	20	3	8	17 1/16
40	14¾	14	19¾	22⅝	17⅞	18	16	14	11¾	22¾	3	8	18 15/16
45	16½	15⅞	22⅜	25⅝	20⅛	20	17⅞	15¼	12¼	25¾	4	9	20 11/16
50	18⅜	17½	24½	28	22	22	19½	16¼	13¼	28½	4	9	22 9/16
60	22	21	29½	33¾	26¼	26	23	19½	16	34¼	5	11	26¼
70	25¾	24½	34½	39½	30½	30	26½	22	18⅛	39¾	6	12	30
80	29½	28	39½	45¼	34¾	34½	29¼	24½	20⅜	45½	6	14	34¾
90	33	31½	44¼	50⅝	38⅞	38½	32¼	27¼	22¾	51¼	7	16	38¼
110	40½	38½	54	61¾	47¼	47	38⅞	32¼	27½	62½	8	20	46¾
130	47¾	45½	64	73¼	55¾	55	45⅜	37½	32¼	73½	9	24	54
150	55	52½	73¾	84⅜	64⅛	64	52⅜	43	37¼	84¾	11	28	63¼
170	62½	59½	83¾	95⅞	72⅝	72	58⅜	49	42½	96	13	32	70⅞
190	69¾	66½	93½	107	81	82	66⅜	54	46⅞	107	15	36	82⅛
200	73½	70	98½	112¾	85¼	86	69⅜	56½	48¾	112¾	16	38	85⅞

DIMENSIONS OF STURTEVANT MULTIVANE FAN

Size	A	B	C	D	E	F	G	H	I	J	K	M	N	O	P
10	40	30½	45	52¼	40¾	37½	26½	36⅜	38⅝	32¾	48¾	32¼	10½	20	66⅝
11	45	34⅜	50⅝	58¾	45½	41½	29½	39⅞	42¾	36¾	54¾	36¼	12	22	73⅜
12	49⅞	38⅛	56⅛	65⅛	51⅛	46	33	43⅛	46¼	40¾	60¾	40¼	12	24	81⅝
13	54¾	42	61¾	71⅝	55⅞	51	36	46⅞	51	44¾	66¾	44¼	13½	28	89¼
14	59⅝	45¾	67½	78⅜	60⅝	55	39	49⅜	54⅜	48¾	72¾	48¼	13½	32	96
15	64½	49⅝	73⅛	84⅞	65⅜	60	43	54⅞	59¾	52¾	78¾	52¼	14½	36	105
16	69½	53⅜	78⅝	91¼	70	64	45	58⅜	64½	56¾	85	56¼	14½	42	112¾
17	74½	57⅛	84⅛	97⅝	74⅝	68	48	62	69⅜	60⅞	91	60¼	14½	48	120¼

The amount of coil surface used in the rooms of a storehouse should be figured by the usual formula:

$$Q = FK(t_r - t_c); \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Q = amount of heat removed per hour in B.t.u.;
F = area of surface square feet;
K = constant of transmission B.t.u. per square foot per hour per degree;
= 5 to 10 for brine or direct-expansion coils to air;
t_c = mean temperature of brine or ammonia, deg. F.;
t_r = temperature of room;
$t_r - t_c$ = 10° to 15° F.

The quantity Q is fixed by the heat entering through the walls and the heat gained by lights, motors, persons and goods stored. The heat from the walls and other causes is computed by methods of Chapter V. The heat given up by articles is given by

$$Q = \frac{Q'}{h} = \frac{M}{h}[c(T_a - T_r) + l_f] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

c = specific heat;
T_a = temperature of articles before storing;
T_r = room temperature;
l_f = latent heat of fusion;
M = weight in pounds;
h = hours to cool and freeze.

Before computing this, h and T must be assumed for any substance. h is fixed by the designer, and the temperature of the room is given in Chapter V. The time h may be taken as from six to twenty-four hours. In all cases it is better to chill slowly. By adding the various heat quantities the total is found.

On account of the ice formation over the pipe the value of K cannot be told exactly, and for that reason the usual method is to allow a number of cubic feet of space for each lineal foot 2-in. pipe for direct expansion or brine.

In some cases 1 ft. of 2-in. brine pipe is allowed to 12 cu.ft. for room temperature of 30° F. while 6 cu.ft. only is used for temperatures of from 5° F. to 10° F.

The following based on Levey's tables may be used for 2-in. pipe and direct expansion in rooms with good insulation, say 1.5 to 2 B.t.u. per square foot per twenty-four hours.

Room Temp.	Cu.ft. Per Foot of 2-in. Pipe.						Limit of Length, Feet.	
	Small Rooms, 1000 Cu.ft.		Medium Rooms, 5000 Cu.ft.		Large Rooms, 10,000 Cu.ft.			
	Brine.	Dir. Ex.	Brine.	Dir. Ex.	Brine.	Dir. Ex.	Brine.	Dir. Ex.
0	1	1	2	2	3	3	100	2000
10	4	5	6	8	8	12	175	2000
20	6	8	10	13	13	19	225	2000
30	8	11	14	17	17	25	275	2000
36	10	14	16	20	20	30	300	2000

The length of the brine coil is limited by the amount of brine which may be cooled and the velocity of the brine, while with direct expansion it is merely a matter of the ammonia which may pass through. The length of the pipes should change with various diameters, since the surface varies as the diameter, while the quantity of brine or ammonia varies as the square of the diameter. The brine coils, however, are not varied in length, as the diameter is changed while the direct expansion-coils vary as the diameter, a 1-in. coil having 1000 ft. as its limit of length. The cubic feet of space cooled per foot of length will vary as the diameter of the pipe.

The allowances made by Louis M. Schmidt are as follows:

DIRECT-EXPANSION PIPING

Freezing and brine tanks	50	sq.ft. per ton of refrigeration.
Brine coolers	10	" " "
Freezing chambers	350	sq.ft. per 1000 cu.ft.
Storage rooms	35	" "

BRINE PIPING

Freezing chambers	500	sq.ft. per 1000 cu.ft.
Storage rooms	50	" "
Bunker rooms	20	" "
Skating rinks	0.8	sq.ft. per sq.ft. of ice surface.

These two sets of tables are practical rules for determining the amount of surface. The better method is to use the formulæ of Chapter V and from them compute the heat lost and the pipe surface to care for the installation, using these tables as checks.

E. H. Peterson in *Ice and Refrigeration* for November, 1915, gives curves showing data for refrigeration of rooms of storehouses. For rooms at zero degrees his curve is to be given by the following equation:

$$\text{Cubic feet per ton of refrigeration} = 1000\left(\frac{\text{cu.ft. room}}{5000}\right)^{\frac{1}{3}}.$$

Increase this by 80% for 10° F., 150% for 20° F., 250% for 30° and 333% for 40°.

One ton will care for 3000 cu.ft. on an average, and 1100 sq.ft. of insulated surface.

If it is desired to find F for the coil surface in a bunker, the quantity K will depend on the velocity of the air over the pipes. This is given by

$$\left.\begin{aligned} K &= 2.2\sqrt{w} \text{ (for wet pipes and wet vapor)} \\ K &= 1+1.3\sqrt{w} \text{ (for dry pipes and wet vapor)} \end{aligned}\right\} \quad (14)$$

w = velocity of air in feet per second;

t_r is then equal to the mean temperature of the air.

The necessary length of pipe is then found and arranged as in Fig. 130, which shows a bunker for air cooling.

It has been found by Sibley that ice formation on the pipe does not cut down the heat transfer, but aids it. An inch of ice in a brine tank coil serves to increase the transfer from a $1\frac{1}{4}$-in. pipe to about four times its previous value, while 2 ins. increases it twelve times. The ordinary allowance of 120 to 150 lin.ft. of $1\frac{1}{4}$-in. pipe of expansion coil in a brine tank per ton of capacity is decreased by him to 40 ft. when ice is allowed to form on the pipes. For ice making this ordinary allowance is doubled to from 240 to 300 lin.ft. Sibley states that when ice can form around the expansion coils in the brine tank the rate of transfer is increased as stated above.

The **weight of ammonia** per hour for a room or plant is given by

$$\frac{Q}{i_1 - i_4} = M_a. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Q = heat in B.t.u. per hour;
M_a = weight of ammonia per hour;
i_1 = heat content at expansion pressure leaving coils;
i_4 = heat content leaving condenser.

The **lines carrying liquid ammonia** should be of such a size that the velocity of the liquid is not over 4 ft. per second.

The **return lines** should be such that the vapor returning is not moving faster than 50 to 100 ft. per second.

These lines may be figured from drop in pressure using the steam formula:

$$P = 0.00015\left(1 + \frac{3.6}{d}\right)\frac{M^2 L}{D d^5}. \quad . \quad . \quad . \quad . \quad . \quad (16)$$

P = pressure drop in pounds per square inch in length L;
L = length in feet;
D = weight of 1 cu.ft.;
d = diameter in inches;
M = weight in pounds per minute.

From this a drop may be assumed and d found or with a given d, P may be found.

Brine is formed by allowing water to flow over calcium chloride or sodium chloride. This is best done by having a box or tank into which the brine is pumped and allowed to flow over lumps of the chloride, and after dissolving some of the salt it is allowed to pass out through the bottom of the tank. The density of the brine is fixed by the temperature at which it is desired to carry the brine. In most cases of closed brine systems it is necessary to keep the brine from freezing, although in congealing tanks it is desired that the brine freeze. The **densities and temperatures of freezing** are given as follows:

FREEZING TEMPERATURE OF SALT SOLUTIONS

Density of calcium chloride { Sp.gr.	1.035	1.062	1.085	1.113	1.122	1.139	1.155
Deg. Beaumé at 64°..	5.0	8.6	11.4	14.7	15.9	17.9	19.7
Density of sodium chloride { Sp.gr.	1.022	1.044	1.067	1.091	1.117	1.142	1.168
Deg. Beaumé at 64.°.	3.0	6.0	9.0	12.0	15.2	18.0	20.9
Temperature of freezing, deg.....	28	24	20	16	12	8	4

Density of calcium chloride { Sp.gr.	1.165	1.174	1.186	1.197	1.207	1.216
Deg. Beaumé at 64°..	20.7	21.7	23.0	24.1	25.2	26.1
Density of sodium chloride { Sp.gr.	1.193					
Deg. Beaumé at 64°..	23.5					
Temperature of freezing, deg.....	0	−4	−8	−12	−16	−20

The **specific heat of brine** varies with the density and the specific gravity. The latest data on this calcium chloride brine are given by Dickinson, Mueller, and George in the bulletins of the Bureau of Standards, U. S. Dept. of Interior, Vol. 6, No. 3, or Reprint No. 135. In this the specific heat of brine at 0° C. is given by

$$D = 2.8821 - 3.6272\partial + 1.7794\partial^2; \quad \ldots\ldots (17)$$

∂ = specific heat in B.t.u. per pound per degree;

D = specific gravity of solution at 0° C. compared with water at max. density.

For changes in temperature from 0° C. the value of ∂ is decreased approximately 0.0008 for each degree centigrade below 0° C.

The specific heat of **sodium chloride** is taken as 0.78 for 1.2 sp.gr., 0.86 for 1.1 sp.gr., 0.94 for 1.05 and 0.98 for 1.02

sp.gr. Having the amount of heat per hour required for a plant or room, the **amount of brine** required per hour is given by

$$Q = M_v \, \partial_m (t_0 - t_i); \qquad \ldots\ldots (18)$$

Q = heat in B.t.u. per hour;
M_v = weight of brine per hour;
∂_m = mean specific heat at temp. $\frac{t_0 + t_i}{2}$
t_0 = temperature of brine at outlet = temp. of room − 10° F.;
t_i = temperature of brine at inlet = $t_0 - 5°$ to 8°.

The length of pipe in one coil is such that this heat of the brine is given up in the length. Thus

$$L = \frac{F}{2\pi d} = \frac{Q}{K\left(t_r - \frac{t_0 + t_i}{2}\right) \times 2\pi d}. \qquad \ldots\ldots (19)$$

The **kind of brine** to use depends on the engineer. Some feel that calcium chloride will not corrode nor rust the iron as fast as the sodium chloride; neither should corrode the piping. On account of the impurities in the salt, brine may corrode the iron. If there is any acid in the brine, corrosion may occur. If there are dissimilar metals in the system, these will set up galvanic action and thus corrosion. Stray electric currents may also start corrosion. In the system there should be no brass pumps if the mains are of iron or steel. It is well to keep the brine alkaline by the addition of lime or dilute caustic soda. The calcium chloride permits of a lower temperature of brine for a given concentration and for that reason it may be employed.

The brine is usually forced through the system by pumps of the direct-acting type, although centrifugal pumps are employed at times. The direct-acting pumps should be of such a displacement that they will deliver their full capacity in cubic feet at 45 cycles per minute.

After the quantity of brine required per hour is obtained by (18), the size of pump is found and after this the pipes carry-

ing the brine should be made of such a diameter that the velocity will be 4 ft. per second. The friction loss is found by using the equation:

$$H = \text{head in feet} = \Sigma f \frac{L}{d} \frac{w^2}{2g} + \Sigma 0.2n \frac{L'}{d'} \frac{w^2}{2g}. \quad . \quad . \quad (20)$$

Σ = summation sign;
L = length in feet of any size pipe;
d = diameter in feet of any size pipe;
w = velocity in feet per second;
n = number of elbows of any size pipe;
L' and d' = dimensions of elbow;

$$f = \text{coefficient} = \frac{0.025}{(dw)^{\frac{1}{4}}};$$

$$\text{Pumping work per hour} = M_v \times H. \quad . \quad . \quad (21)$$

Cold-storage warehouses are sometimes operated from a **central refrigerating plant.** These central stations will pay when there are a number of persons needing refrigeration within a limited radius. In the warehouse districts of Boston, New York, Philadelphia, Baltimore, Norfolk, St. Louis, Kansas City, Denver and Los Angeles and in the hotel district of Atlantic City central stations have been installed. The lengths of mains vary from 1 to 17 miles and the income amounts to about $12,000 per mile. The **systems may be of the brine system or the direct-expansion system.** In each case there must be at least two mains, a supply and a return. **In the brine system** the pipes must be carefully insulated, as the brine is at low temperature and about $2\frac{1}{4}$ H.P. is required by the brine pump motor per ton of refrigeration to drive the brine through the main. The pipes are put in wooden boxes after covering them with hair felt soaked in rosin and paraffin oil. The box is waterproofed. The arrangement of the box is shown in Fig. 133. In this one set of pipes is arranged in a wooden box while in the other a split-tile conduit is used. The pipe is carried on supports at 12-ft. intervals for 2-in. pipe and over, while for 1-in. pipe 8-ft. intervals are allowed. At certain points the pipe line is anchored and on each side of this anchor

expansion is allowed to take place. Expansion amounts to 0.08 in. per 100 ft. for each 10° of temperature change. Expansion joints are placed at about 175 ft. intervals and should be of the pipe bend or swinging ell type, although the slip joint as shown in Fig. 134 is used. The anchor points should be

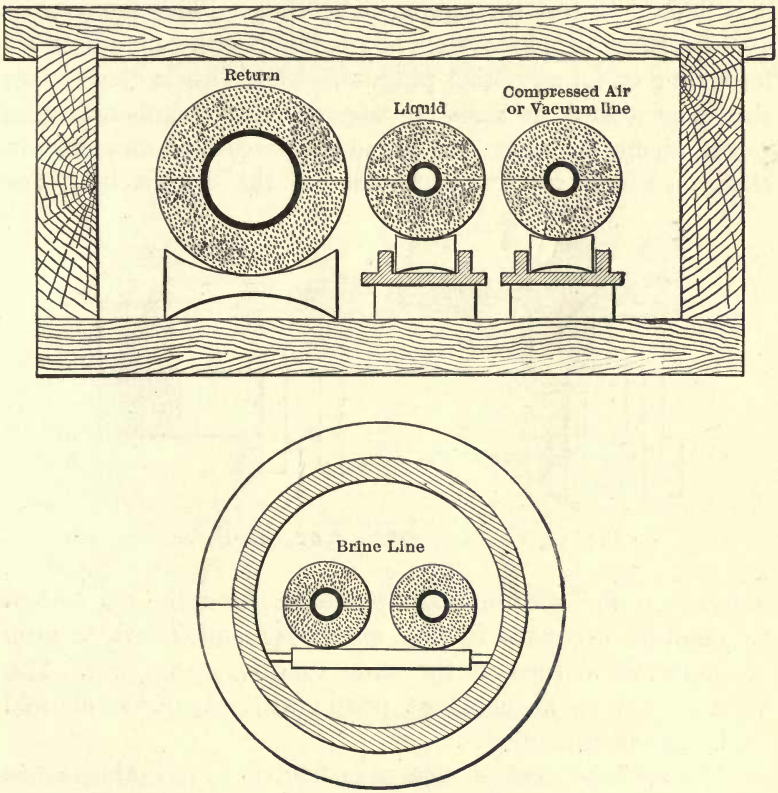


FIG. 133.—Conduits for Refrigerating Pipes.

points at which branches are taken off. The pipes used for brine may be of cast or wrought iron.

In the **direct-expansion system** there is no need of insulating the supply main, as this will not absorb heat from anything as cool as or cooler than the cooling water, since the ammonia is under pressure. The return main should be in-

sulated, although this is not necessary if the ammonia vapor is warmed by the abstraction of heat in the storehouse to at least earth temperature. In direct-expansion installations it is customary to run three mains with cross-connections at manholes and warehouses. One pipe is used as the pressure main, one as the return and a third as a vacuum line to be used when it is desired to test the piping in buildings. This line can be used to charge the pipe system of a building with compressed air for testing or for any other purpose. The joints in this system should be welded, as leaks are very expensive, ammonia being worth about 25 cents per pound. The cost of ammonia to charge such a system is another of the drawbacks. The

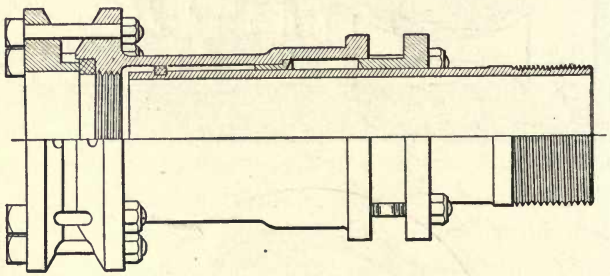


FIG. 134.—Expansion Joint for Ammonia Line.

pressure in the suction main would be fixed by the coldest temperature necessary in any warehouse, and usually a drop of 15 lbs. is allowed in the main to drive the vapor. The main should be anchored at points with expansion allowed for by bends in manholes.

The **load for such a station** is figured by allowing 1 ton for about 3000 cu.ft. for spaces up to 40,000 cu.ft. For insulated areas an allowance of 1100 sq.ft. to the ton will care for walls, floor and ceiling. The temperature on the three hottest consecutive days is used in computing the peak load. These may be found by getting the records of the nearest weather bureau office.

For **brine lines** bell and spigot cast-iron pipe has been used. Voorhees has installed 1500 ft. of 10-in. pipe of this kind and is

supplying two warehouses of a total capacity of 1,500,000 cu.ft. He could not detect a rise of temperature in this length on thermometers reading 2° to $\frac{1}{8}''$, showing that the gain of heat in the duct was not great.

The **pipes should be installed perfectly dry** and kept that way. Paint is of little value when pipes become wet. The best thing to use as a paint is some form of bituminastic solution.

Automatic Refrigeration

The use of automatic apparatus by which the temperature of rooms is kept constant is one of the recent develop-

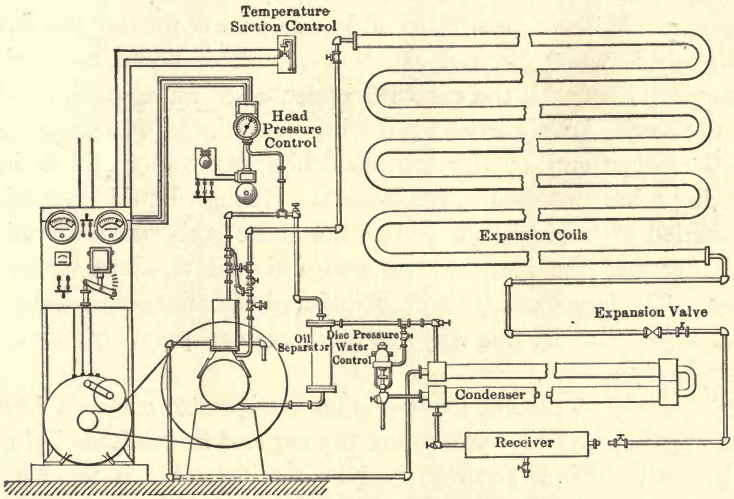


Fig. 135.—Automatic Refrigerating Plant.

ments of the art. In this method an electric device controls the expansion valve and the pressure in the expansion coil regulates the motor operating the compressor. In a similar way the pressure in the discharge main controls the water supply to the condenser and on the pressure reaching a limiting high value the apparatus is shut down, thus guarding against the failure of the condensing water supply.

The claim for such a machine is that the amount of refrig-

eration is just that which is needed. It will be easily understood that when a room is cooled off below the required temperature the heat flow is increased and more work than that necessary for the plant is done. In automatic installations a constant temperature may be maintained and one no lower than that required. Of course this also prevents the temperature from rising above the desired point, and although this would rarely happen in a well-operated plant, a much lower temperature may be carried to prevent it.

To obviate the necessity of **charging refrigerator cars** with ice, compressor plants have been proposed. For instance in *Ice and Refrigeration* for May, 1910, there is a description of a patented system in which a small compressor is placed on the car. A device exhibited in Paris in 1900 for the Russian railroads is worth noting. In this a tank of liquid ammonia was placed beneath the car and connected to an expansion coil in the car. An absorber filled with weak liquid was attached to the other end of the coil, absorbing the vapor and maintaining a low pressure in the system. Enough liquid ammonia is carried for a given run and at the end of this run the liquid cylinder and the absorber are removed and replaced by new ones. The liquor can then be boiled and the ammonia regained and liquefied. In this way the most remote points of the car may be kept cool.

The cold storage of foodstuffs on refrigerator cars has been recently improved by **precooling the car and its contents** before shipment. This is accomplished by forcing cold air in at one end of the car and drawing out warm air from the other. After a given length of time the current is reversed and air is drawn from the end into which it was forced, and withdrawn from the other end. In this way a large amount of heat can be taken from the car before ice is introduced.

There are several of these **precooling stations.** The Santa Fe System has one at San Bernardino, California, and the Southern Pacific, one at Roseville, California. At these stations twenty-four to thirty-two cars are placed in connection with ducts which are covered with a heat insulator. By movable,

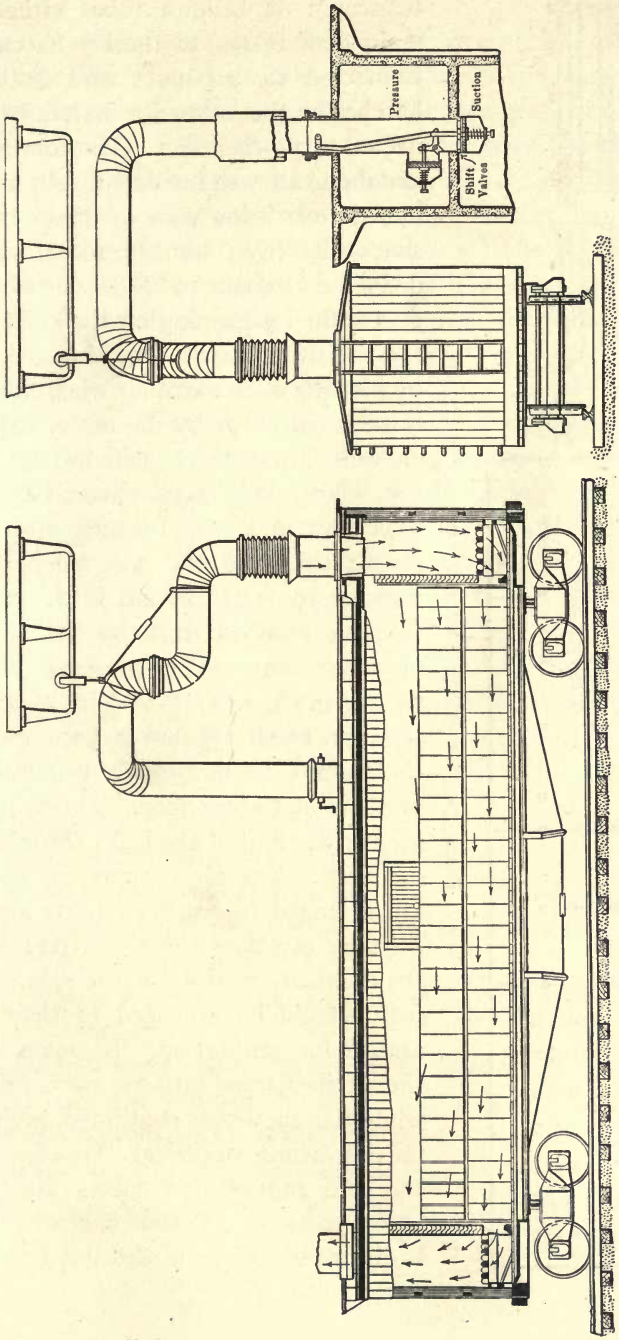


FIG. 136.—Arrangement of Car at San Bernardino.

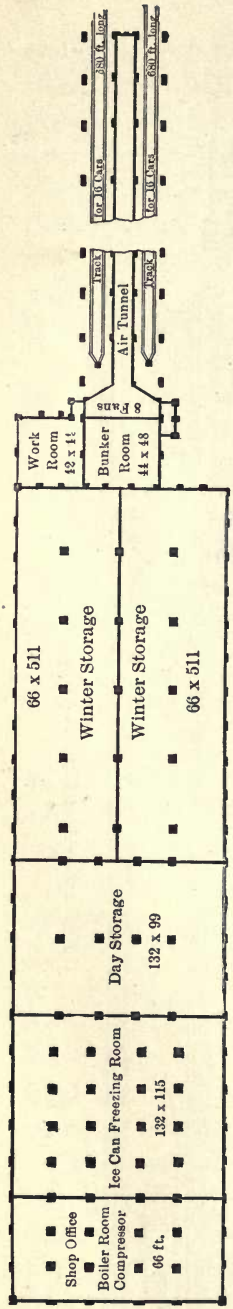


FIG. 137.—Plan of San Bernardino Pre-cooling Station.

telescopic or bellows tubes either the main door or one of the ice hatches is connected to one duct and both ice hatches or the other ice hatch is connected to another duct. The connection and ducts are well insulated. Air is then blown over brine coils or direct-expansion coils in a bunker room and is delivered at about 10° F. to one of these ducts, the ice forming on the coils from the moisture in the air being removed by blowing over warm air when the cold brine is cut off or by the use of calcium chloride brine which trickles over the coil. The air is then delivered by fans to a duct and then by means of telescopic tubes to the car, where it is warmed 20 or 25° F. and is drawn out through another duct by the suction of other fans. The pressure of the forcing fan is from $\frac{1}{2}$ to $\frac{3}{4}$ in. of water, while an equal vacuum is produced by the suction fans giving atmospheric pressure at the car door. This is necessary on account of the leaks through the car walls. The air currents in the car are arranged to reach all parts and by a system of valves the air currents from the ducts are reversed in the cars. The fruit should be arranged in tiers separated for ventilation. In some cases the air discharged into the car is shut off while the suction is continued, giving a partial vacuum in the car. On admitting the cold supply this enters the parts of low pressure. The vacuum also tends to draw out some of the gas from the

fruit. At San Bernardino the rate of 6000 to 8000 cu.ft. per minute per car is used for a period of four hours, reducing the temperature of car and goods to 40° F. The arrangement of the car is shown in Fig. 136. Fig. 137 shows the general arrangement of the plant, which represents an investment of about $900,000.

After these cars are reduced to 40°, or the temperature desired, they are iced at the station and then shipped east. The car may then be sent to Chicago without further icing and the fruit will be in far better condition than when treated with ice in the ordinary way. The cost of refrigeration as ordinarily run with ice from California to Chicago is $62.50 per car, while after precooling and original icing by the shipper the further icing is reduced to $7.50 per car. The cost of precooling is $30.00 per car and the original icing is $25.00.

At Springfield, Mo., the United Fruit Co. cool their cars by this method, lowering the temperature 26° in twenty-four hours. In this way they may be held cold, the rise being 2 or 3° in 500 miles of travel.

At these precooling stations ice is made for charging cars, and for that reason the plants are equipped with ice storage rooms. A storage plant is necessary for the cooling of the cars and the manufacture of ice. The plant equipment is as follows:

Santa Fe at San Bernardino, Cal.:
- Ice-making capacity: 225 tons per day.
- Ice storage: 30,000 tons of ice (day room 900 tons).
- Compressors: two 300-ton Vilter refrigerating machines.
- Cars at one setting: 32.
- Bunker rooms: 44 ft. 6 ins. by 48 ft. by 9.3 ins., and 8 ft. 9 ins.
- Fans: eight 120-in. Sirocco fans, 65,000 cu.ft. per minute each at $\frac{3}{4}$ in.
- Insulation: 3-in. cork on concrete.

Pacific Fruit Express Co., Roseville, Cal. (S. Pac. Co.):
- Ice-making capacity: 250 tons per day.
- Ice storage: 20,000 tons (112 by 115 by 32 ft. and 75 by 115 by 32 ft).

Cars at one setting: 24.
Bunker rooms: 80 by 26 by 9 ft. and 8 ft. high.
Compressors: two 250-ton York compressors.
Insulation ice house: concrete walls with 3-in. lith.
Air duct 1-in. boards: $\frac{1}{2}$-in. asphalt, 1-in. boards, $3\frac{1}{8}$-in. granulated cork, 1-in. boards.
Fans: four fans 85 in. diameter, $27\frac{3}{4}$ in. wide, 44,500 cu.ft. per min. at 380 R.P.M. Pressure 3 oz.

That this is not an untried invention is shown by the fact that over 22,000 cars have been precooled at the Santa Fe plant alone during a period of about five years.

CHAPTER VII

ICE MAKING

In making ice two methods have been used for a number of years. In one, distilled water is placed in cans and these cans are surrounded by brine, liquid ammonia, or cold air, which removes heat from the water and causes it to freeze. In the other, water taken from a stream or other supply (known therefore as **raw water**) is placed in a large tank which contains a number of coils of pipe for vaporizing ammonia or circulating brine. This removes heat from the water until gradually a plate of ice is formed on each face of the coil. If a large tank, 20 ft. wide by 60 ft. long by 8 ft. deep, contain 20 coils, 40 plates of ice could form. The first method is known as the **can system**; the second as the **plate system.**

To study the peculiar details of these systems two general plans will be examined, after which the details of construction and operation will be considered.

Figs. 138 and 139 are the plan and elevation of a 25-ton standard can system of the York Mfg. Co. In the plan view, Fig. 138, the compressed ammonia is delivered from the compressor to a double-pipe condenser, passing through a separator before entering the condenser. From this point the liquid ammonia is carried to an ammonia receiver shown dotted and seen in Fig. 140, and it is then taken to a liquid line on top of the brine tank at *A*, Fig. 139. From the liquid line it passes through expansion valves and enters the expansion coils of $1\frac{1}{4}$-in. pipe shown in the longitudinal section, Fig. 139. The liquid enters at the top of the coil and flows down to the bottom, abstracting heat from the brine. From Fig. 140 it will be seen that there is a coil between each two rows of cans. The manifold or pipe line *A* leading the liquid to the coils is carried

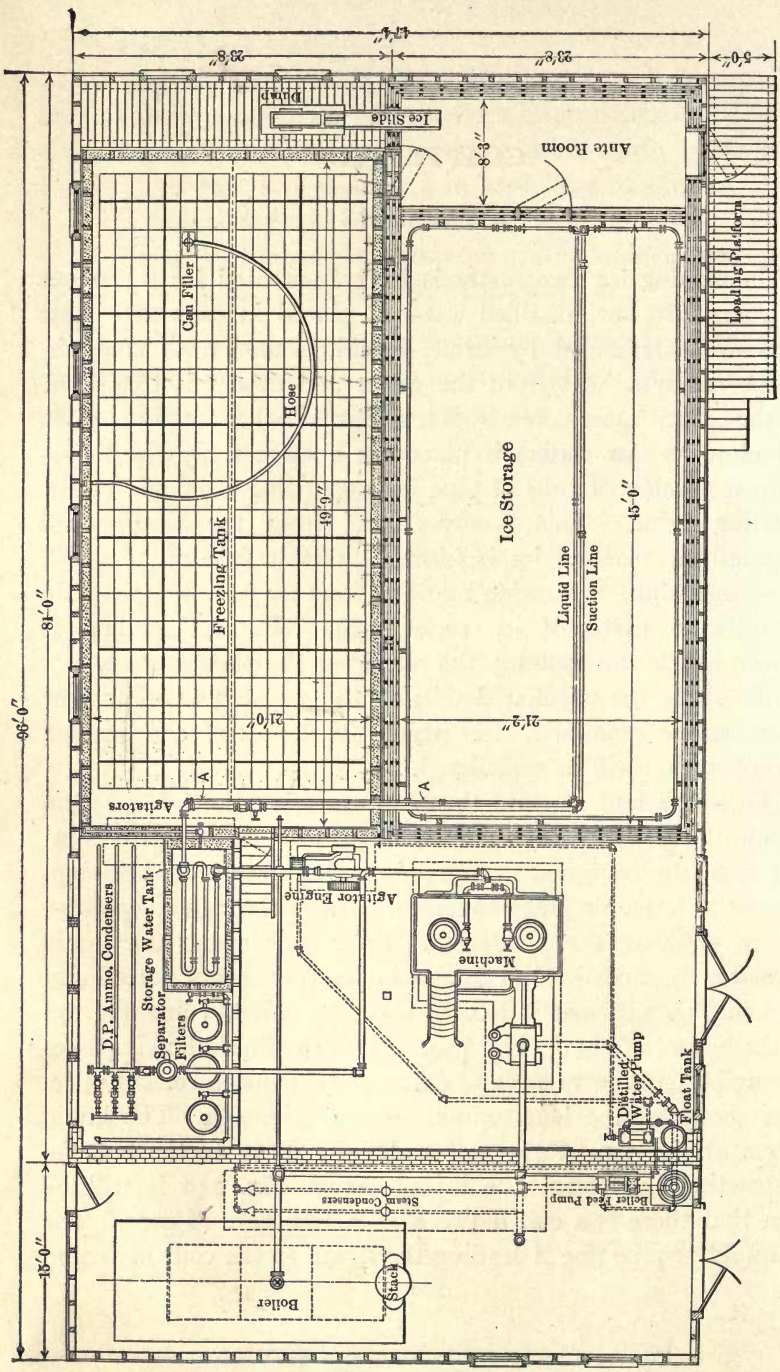


FIG. 138.—York Mfg. Co. 25-ton Ice Plant.

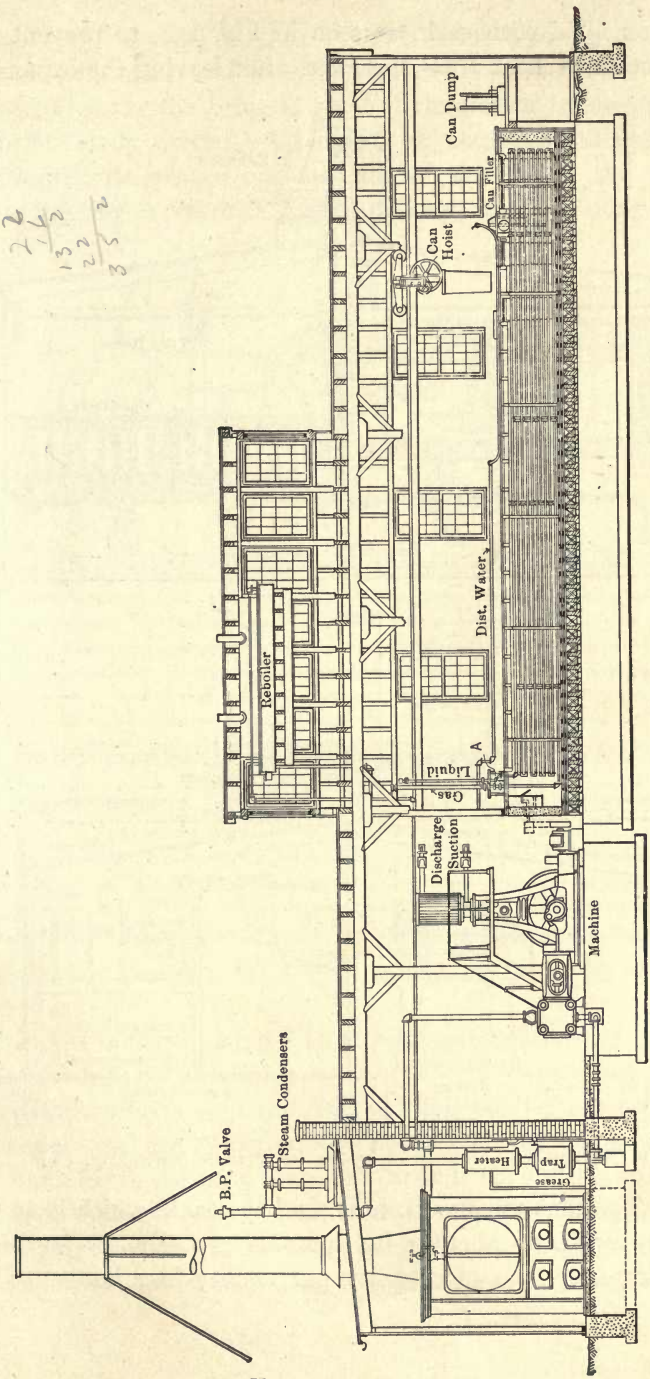


FIG. 139.—Section of York Ice Plant.

in a box filled with cork, as seen in Fig. 139, to prevent the abstraction of heat from the room after leaving the expansion

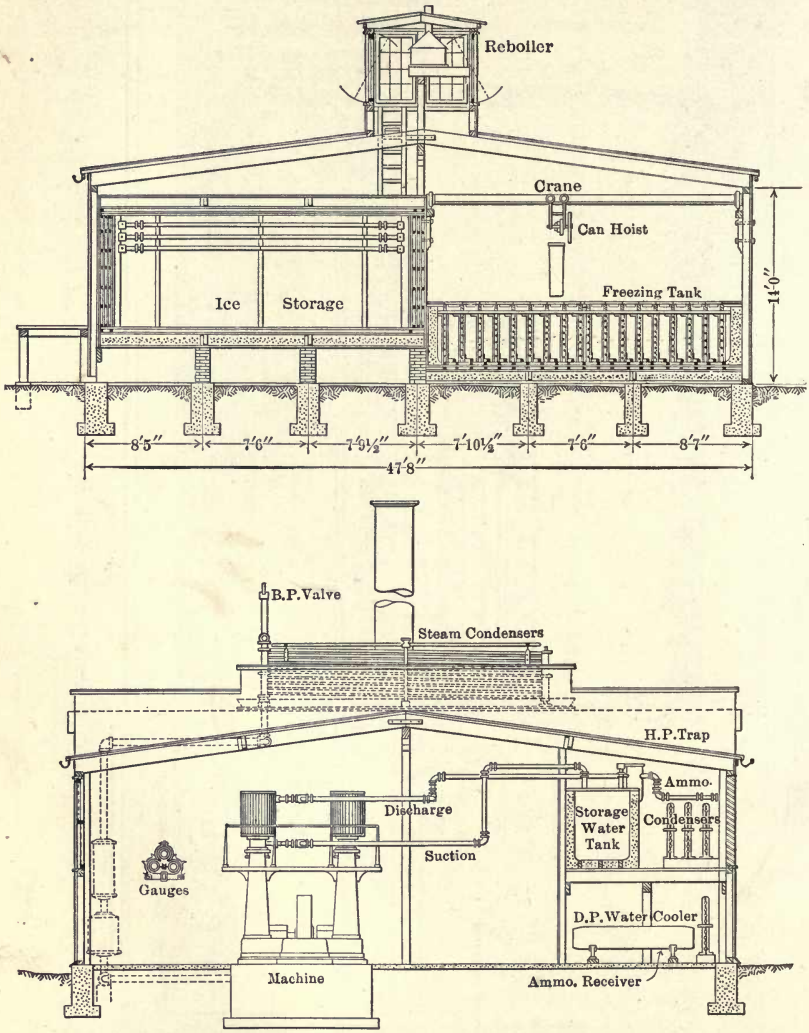


FIG. 140.—Cross Sections of 25-ton Ice Plant. York Mfg. Co.

valve. The cans are 300-lb. cans, and the tank, which is 49 ft. 9 ins. by 21 ft. by about 5 ft., contains 352 cans, or fourteen 300-lb. cans per ton of capacity per twenty-four hours. The

cans are $11\frac{1}{2}''$ by $22\frac{1}{2}''$ by $44''$. The **brine-freezing tank** is made of $\frac{1}{4}$-in. steel and is well insulated on the sides. As shown in Fig. 141 the brine is given a **circulation** by means of a propeller blade driven by a motor or belt from an engine. This propeller is placed on one side of the end of the tank and by running a vertical partition longitudinally along the

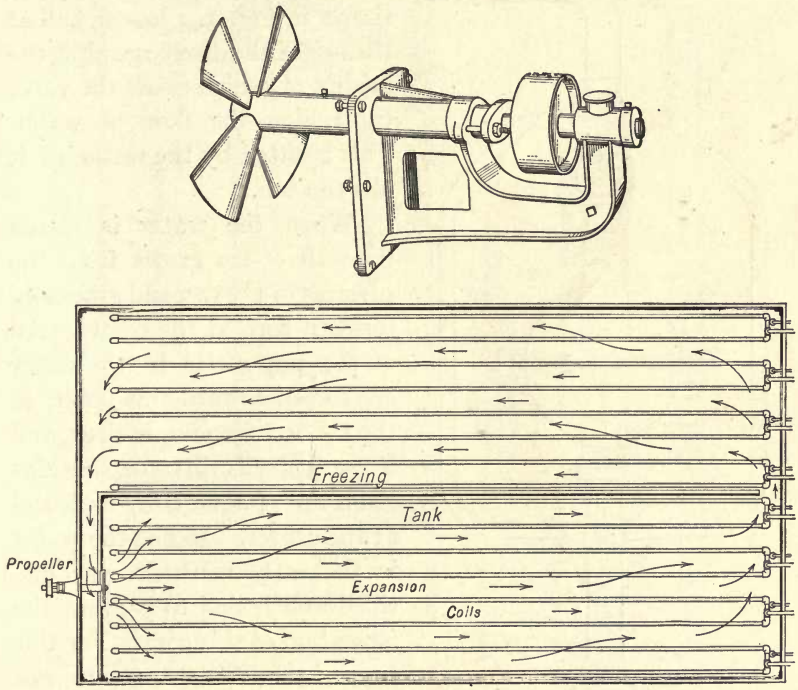


FIG. 141.—Plan of De la Vergne Freezing Tank with Brine Agitator.

center of the tank to within 2 ft. of each end a channel is made to cause a definite circulation.

The vapors formed in the coils are collected in a **return or gas header** and are taken to the **suction of the compressor.** This line, Figs. 138 and 140, is carried through a **storage water tank** so as to remove heat from the distilled water which is to be placed in the cans. This completes the passage of the ammonia and the compressor delivers the ammonia again

to the condenser. The water from the storage tank is taken to a point on the outside wall of the center of the freezing tank length and then discharged by a hose into cans which have just been emptied. An automatic device shown in Fig. 142 cuts off the water supply when the tank fills. These **fillers** are of various forms, differing in detail. The **K–C Filler** shown in Fig. 142 has a ball at the top of the discharge pipe, the raising of which closes the valve controlling the flow of water. This is lifted by the water as it fills the can.

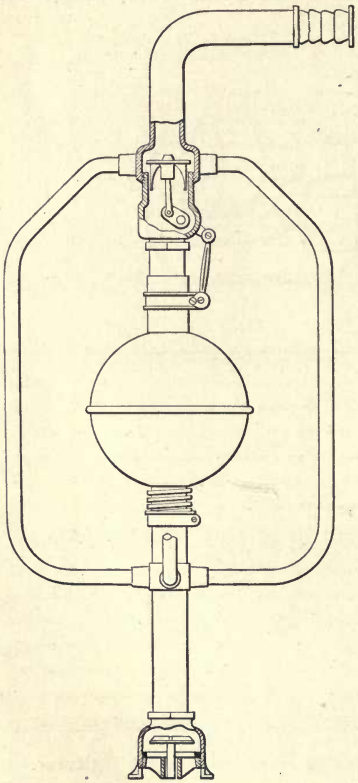


FIG. 142.—K. C. Can Filler.

When the water is frozen the wall of **ice grows** from the surfaces of the can and gradually forms a core at the center. All of the impurities in the water are forced toward this point, as the ice first forming is clear, and if the water is dirty or contains scum an opaque core is found at the center. Hence the water in the early methods was distilled and boiled to prevent the formation of this core. For this reason the exhaust steam was condensed and used. In the plant shown, Figs. 139 and 140, the exhaust from the engine is carried through a grease separator to a condenser placed on the roof of the boiler house. From here the condensed steam is carried to a **reboiler.** In this the water is brought to a boiling temperature by a steam coil and the oil and other impurities remaining in the condensed steam as well as air are forced out.

This reboiler is placed in the **monitor** of the roof. The water

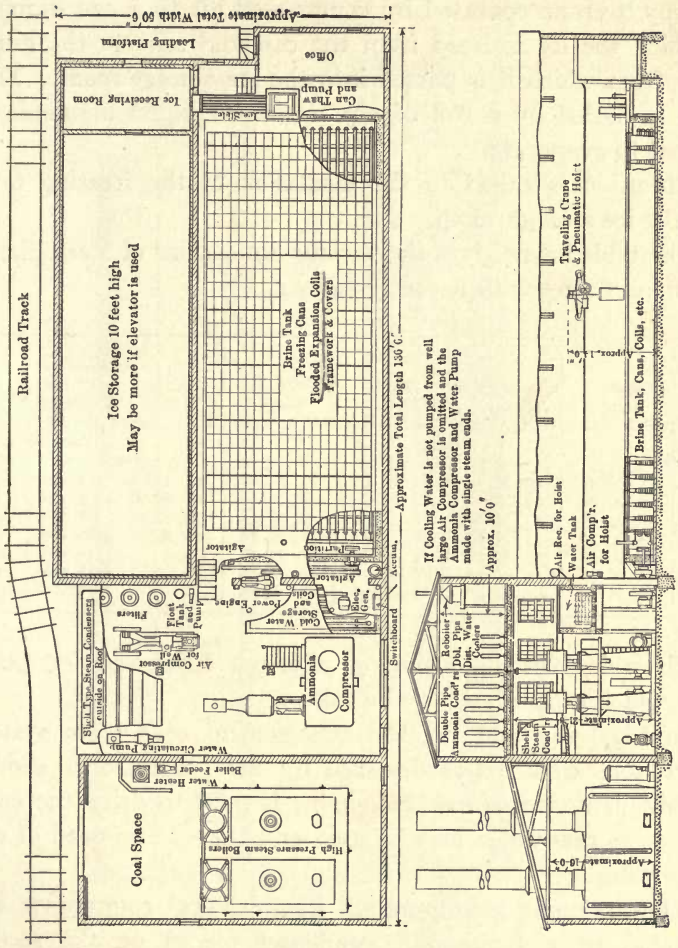


FIG. 143.—Frick Ice Plant.

is then taken through a **cooling coil** and finally passed through several **filters** for cleaning and deodorizing before entering the **storage tank.**

When the water in the can is frozen it is taken from the brine tank by a crane operated by compressed air to a can dumper in which the ice is freed from the can and sent to the **anteroom**, from which it is passed into the **ice-storage room.** This room is cooled by a coil of pipe in which liquid ammonia is allowed to evaporate.

Attention is called to the **insulation** of the freezing tank and the ice-storage room.

The table below gives the various dimensions of York plants of different sizes with necessary data:

Tons.	Boiler Room. Wdth. Lgth.	Compression Room. Wdth. Lgth.	Tank Room. Wdth. Lgth.	Storage Room. Wdth. Lgth.	Ante Room. Wdth. Lgth.	Lin.ft. of $1\frac{1}{4}$" Exp. Coil in Brine Tank.	300-lb. Cans.
10	15 × 40	25 × 40	20 × 35	20 × 27	20 × 8	2,500	160
20	15 × 44	25 × 44	22 × 50	22 × 40	22 × 10	5 000	320
30	18 × 56	25 × 56	28 × 53	28 × 43	28 × 10	7,500	480
40	18 × 56	25 × 56	28 × 82	28 × 60	28 × 12	10,000	640
50	18 × 56	25 × 56	28 × 97	28 × 97	.....	12,500	800
75	18 × 56	25 × 56	28 ×128	28 ×128	.....	18,750	1200
100	18 × 56	25 × 56	28 ×158	28 ×158	.....	25,000	1600

Fig. 143 gives the outline of a 50-ton Frick can ice plant from which dimension may be taken.

In Figs. 144 and 145 the arrangement of a **plate system** for 50 tons capacity as designed by the Frick Co. is shown. In this plant a producer gas engine is used to drive the compressor, as **raw water** may be used and there is no need of distilled water.

The ammonia is compressed in a vertical compressor and delivered to a double-pipe condenser placed on the second floor of the engine house, Fig. 144. From the condenser it is taken into a **receiver** *A* in the freezing room near the wall and then delivered to a **header** *B* running along the freezing tanks. In the figure shown there are seven **tanks** about 20 ft. long, 10 ft. wide and 10 ft. deep, each well insulated against

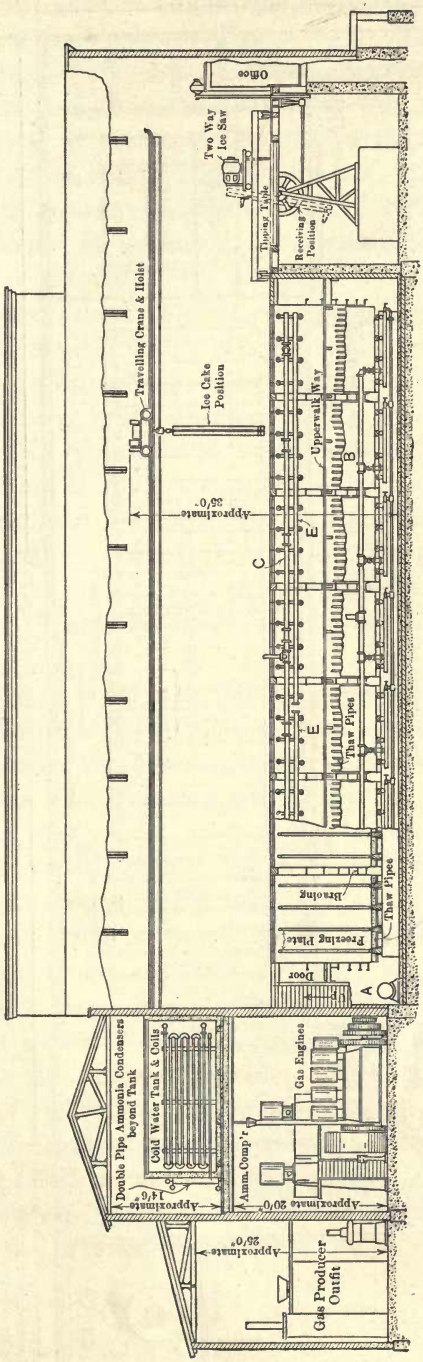


FIG. 144.—Section of Direct-expansion Plate Ice Plant. Frick Co.

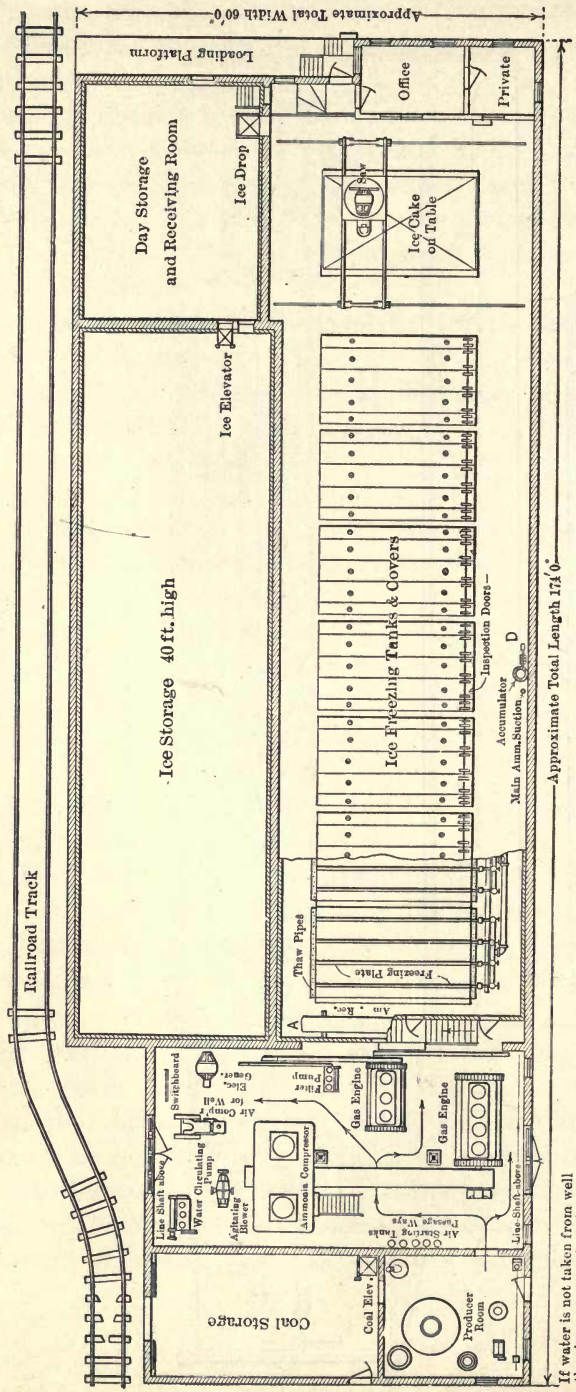


FIG. 145.—Plan of 50-ton Plate Ice Plant. Frick Co.

heat loss. In each of these tanks are four direct-expansion coils supplied from a header connected to the main liquid line. By throttling the ammonia is allowed to enter the coil and the vapor is taken off from the mixture leaving through the main suction pipe *C* at the top. The arrangement of feed is such that there is much liquid left in the coil at the top and the mixture is taken from the top of the coil to an **accumulator** *D* where the vapor is separated, the liquid passing back into the lower part of the coils. On each side of an expansion coil is placed a heavy sheet of steel on which the ice forms. This ice forms gradually, taking six or seven days to make 12 ins. of ice, the plate weighing about 6 or 7 tons. As the ice forms the impurities are forced ahead of the ice, leaving clear ice from raw water. To release the ice when ready to harvest, the liquid is cut off from the coil and warm vapor is allowed to enter from the header *E*, this melts the ice from the plate and steam or warm brine is passed through pipes placed beneath the tank and on the sides to melt the ice from the sides and bottom of the tank. This is the only function of the thaw pipes shown in Fig. 144.

The plate of ice is now lifted by a crane, using iron rods frozen in the ice to carry the load and after placing the plate on a **tilting table** and bringing it to a horizontal position, a **saw** is used to cut the ice into small blocks. The saw is mounted on a motor and moves on a sliding table. The blocks are stored in the anteroom or in the main storage room.

The **tanks are carefully insulated** and the piping beyond the expansion valve is covered to prevent loss of refrigerating power. The tanks are well braced. The arrangement of piping is such that when warm ammonia is introduced from a special line the condensed ammonia may be drawn from the coil by a liquid transfer header. The liquid ammonia could go to one of the coils using ammonia liquid. The size of each tank is sufficient to give the complete tonnage of the plant for one day so that this tank can be emptied of ice and filled with fresh water the same day. The water remaining with the impurities is taken off. This would require that the num-

ber of tanks equal the number of days required to freeze the ice to its desired thickness. The plant shown will make about 50 tons.

With plate ice raw water may be used and there is no need of distilled water. Hence high-grade steam engines, gas engines or electric motors may be used for the prime movers. The use of the electric motor when power is taken from a central station during the off-peak hours of a public service company, furnishes a cheap method of driving, as the off-peak rate may be very low. The Chicago Edison will sell off peak power at 1 cent per kilowatt hour. In the figures shown the plant is driven by gas engines throughout. The water-circulating pump, the agitator blower, the air pump for the deep well, the electric generator and the filter pump are driven by a small gas engine. Air-storage tanks for starting the gas engines are placed on the side of the room. The gas producer and coal-storage room may be seen.

The **cold-water storage tank** is used to cool the raw water after it has been filtered. This filtering is resorted to to take out suspended matter and bacteria.

Having the general arrangement of the two systems, it will be advisable to examine the peculiar apparatus used with each.

The **distilled water apparatus** is of various forms. When there is sufficient steam from the engines an arrangement is used as shown in Fig. 146. This is that used by the York Mfg. Co. The exhaust steam from the engine passes through an **oil separator** and **feed-water heater** and then to a **condenser,** where it is condensed by water used in the ammonia condenser. The line is equipped with a **free exhaust valve** set to relieve the pipe after a certain pressure is reached. The condensed steam is then collected in a return tank, from which it is pumped into the reboiler. This reboiler is operated with steam from the exhaust main or from live steam with the condensate passing to the exhaust line. After reboiling the water is passed to a double-pipe water cooler and is delivered to a storage tank after passing two **filters** and a **regulator.** The cooling may be

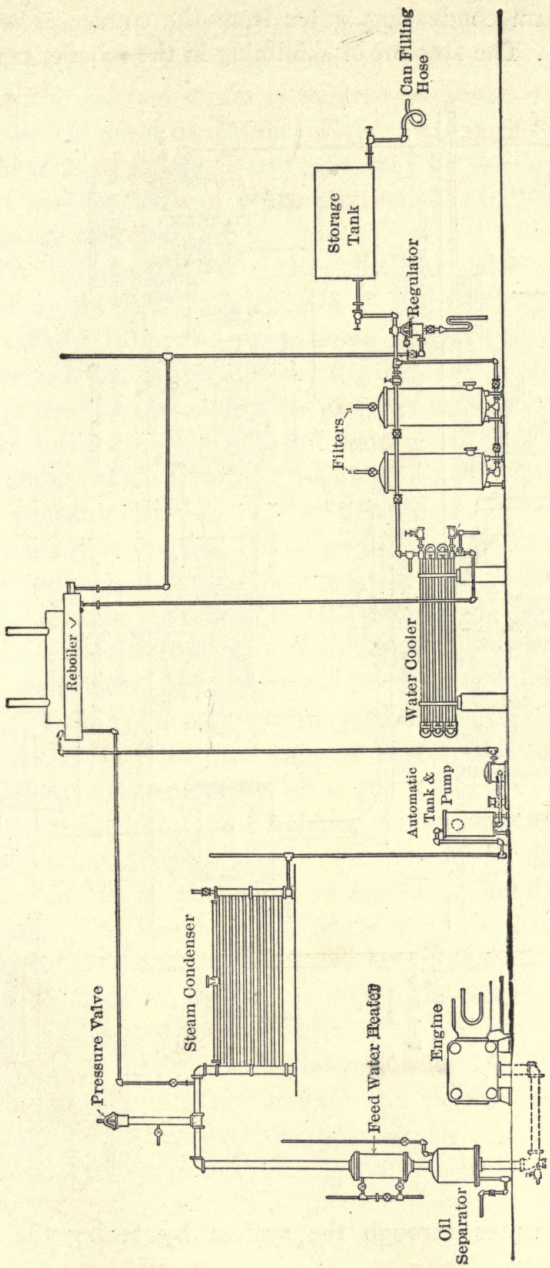


FIG. 146.—Distilled Water System of York Co.

done by warm condensing water from the condenser or by a cool supply. The amount of skimming in the reboiler regulates

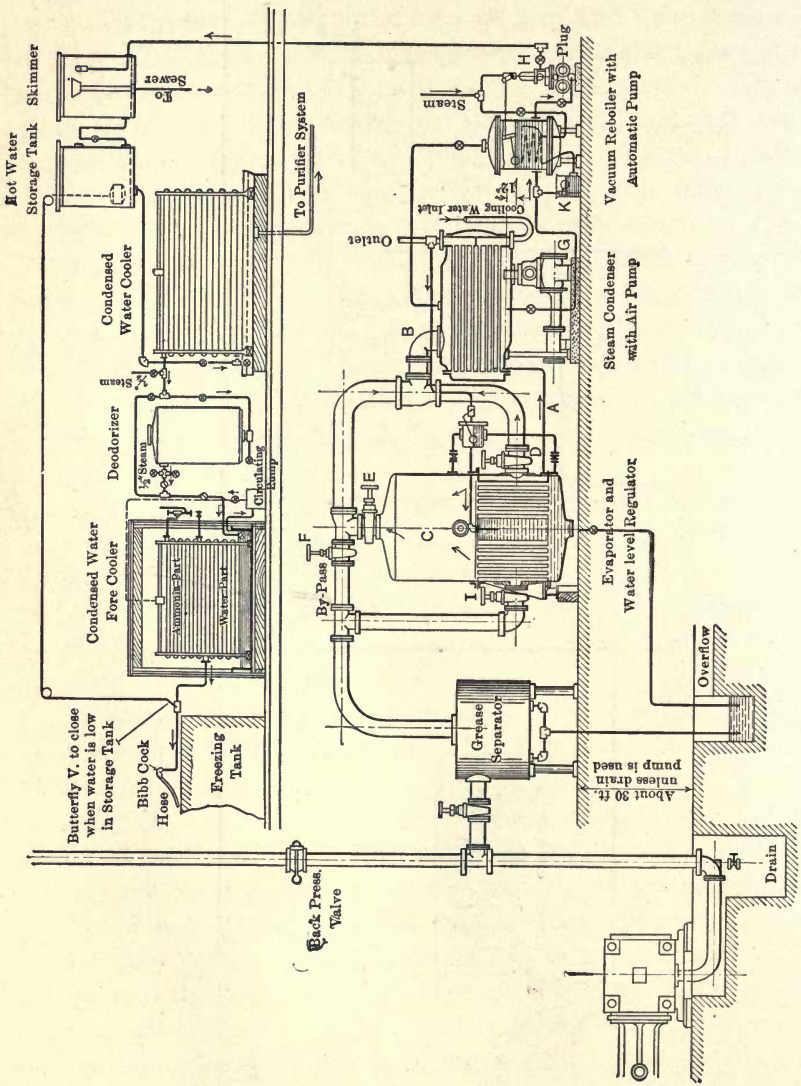


FIG. 147.—De la Vergne Vacuum Distilling Plant with Evaporator.

the flow of water through the system by taking the scum to the regulator.

When the exhaust steam is not sufficient to supply the distilled water required, some form of **evaporator** is installed by which the exhaust steam is used to evaporate water and thus increase the yield of distilled water. At times the **Lillie evaporator** is used, although any type may be employed. A single effect evaporator (one evaporator only) is in general sufficient for an ice plant.

The arrangement of such a distilled water system as proposed by the De La Vergne Co. is given in Fig. 147. In this the exhaust from the engine passes through a pipe containing a free exhaust valve and passing through a grease separator enters the space around a set of tubes in a **vertical evaporator.** These tubes are held between **tube plates** and are filled with **raw water** which rises to a point above the upper tube plate. The condenser *B* has a vacuum maintained by the air-pump at such a pressure that the water in the tubes of the evaporator boils and passes over to the condenser. The boiling of the raw water removes the heat of the exhaust steam and this condenses around the tube and collects on top of the lower tube plate. The condensate is drawn over through pipe *A* to the steam space of the condenser by the vacuum in *B* and is allowed to flow by gravity into the **vacuum reboiler** which is connected at the top to the steam space of the condenser. The **air pump** *G* is connected to the condenser above the water level and removes only air and vapor. The reboiler is freed from water by the pump *H*, which has an **automatic float control** in the reboiler. The reboiler takes live steam from the pump supply and the condensate from the coil is caught in a **reservoir** from which it is drawn by the vacuum in the reboiler whenever the valve *K* is opened by the float.

The exhaust steam discharged from the engine may be by-passed around the evaporator by closing *D*, *I* and *E* and opening *F* and by partially opening *D*; when *F* is partially opened and *I* and *E* are open, some of the engine steam is allowed to flow to the condenser without evaporating any water. In this way the amount of distilled water is regulated.

The **reboiled water** is pumped into the **skimmer** and then

enters the hot-water storage tank and passes successively through the cooler, deodorizer and fore-cooler. The **storage tank** for hot water is provided with a float so that when this water

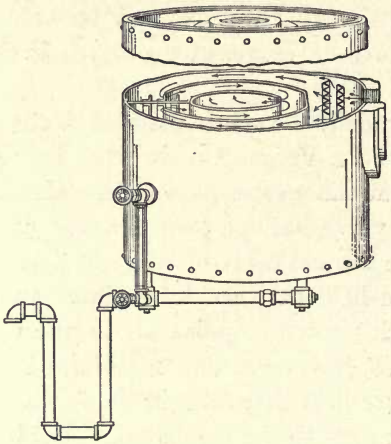


FIG. 148.—De la Vergne Grease Separator.

is low the **butterfly valve** on the final discharge pipe is closed and prevents water from being drawn away. The **atmospheric water cooler** is cooled with water which may be used later for water supply. The **deodorizer** is a charcoal filter to remove

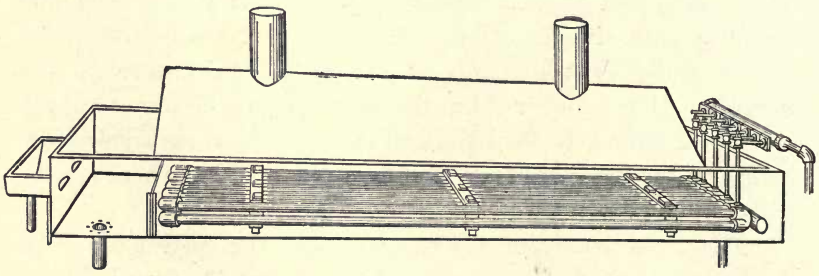


FIG. 149.—York Reboiler.

odor and certain suspended matter. In the **fore-cooler** the water passes from it through a set of pipes which are cooled by ice water flowing over the set. This ice water is pumped from a tray beneath the coil and passed first over pipes which

contain low-temperature ammonia gas or ammonia liquid. In this way the cooling water is cooled and there is no danger of freezing the water within the lower pipes, as the flowing water cannot reach a point below 32 unless it freezes on the outside of the ammonia coils.

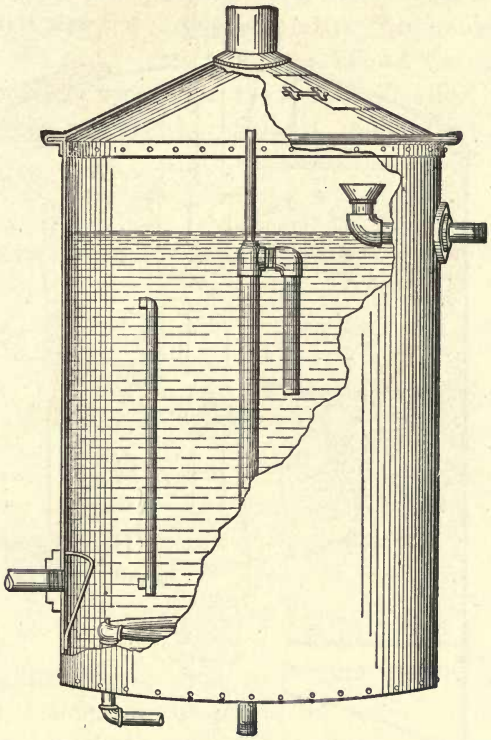


FIG. 150.—De la Vergne Reboiler and Skimmer.

The arrangements in these figures give the general outline of all apparatus for distilled water.

The arrangement of the **grease separator** of the De La Vergne Co. is shown in Fig. 148. The action of the baffle plates on the steam and oil is to remove the oil and water.

The **reboiler** of the York Co. is shown in Fig. 149 while Fig. 150 gives that of the De La Vergne Co. In each of these a steam coil causes the water to boil and thus drives off air or

other gases, while the oil which forms a scum is taken off by the skimming edge of the holes at the right-hand end of the outlet chamber.

The **filters** are vessels containing a layer of quartz sand

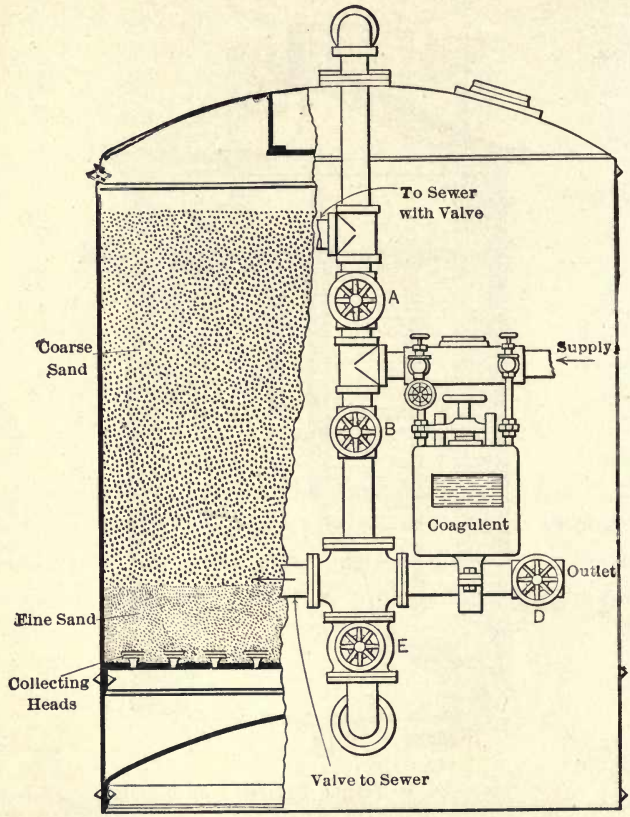


FIG. 151.—Sand Filter.

on top of which is placed charcoal. This removes the last suspended matter as well as the odor and taste, and in order to clean this an occasional supply of steam can be admitted to melt off the oil which may collect. The same can be done to the pipes of the condensed water cooler. See Fig. 147.

When raw water plants are used the **filters** may be of the

sand type as shown in Fig. 151. In this there is a large shell of steel containing a number of **collecting heads** in a lower diaphragm, through which the water passes after traversing a 4-ft. thickness of graduated sand, fine at the top where the water enters and coarser as the bottom is reached. By introducing a small amount of alum solution from the coagulent box into the water in inlet pipe the suspended matter coagulates and collects on top of the bed. This collection, known as the "schmutzdecke," gradually becomes so thick that filtration is slow. The current is then reversed by valves *A* and *B*, the water entering

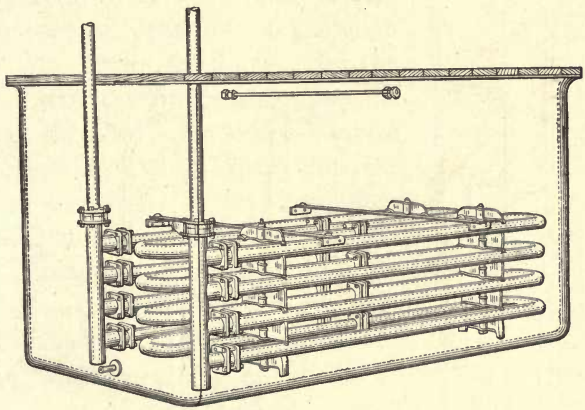


FIG. 152.—Frick Distilled Water Storage Tank.

from the bottom by *E*, while valve *D* is closed and a valve to one side of *A* connects the top to the sewer. In this way the washing removes the deposit from the filter. A glass cup shows the condition of the water passing. After the main washing is completed, the current is reversed to its proper direction and the discharge allowed to waste to the sewer through a valve on the left of *E* until the water is clear, after which it is cut off from the sewer and passed back to the line. The maximum amount of water cared for by such filters with safety is 2.5 gallons per minute per square foot of filter bed.

If a **storage tank** is used in place of the fore-cooler the water is cooled by the circulation of ammonia. The ammonia

is either allowed to evaporate in a coil in the tank or the cool ammonia vapor from the expansion coils is taken through the coil and is warmed by the extraction of heat from the water. In this way ammonia may be superheated on entering the compressor. Fig. 152 is a view of the Frick storage tank with the coil shown as if the tank were transparent.

The construction of the **freezing tank** for the can system of the De La Vergne Co. is shown in Fig. 141. The inner partition makes a circulation from an agitator positive, as shown by the arrows. The coils are arranged so that two

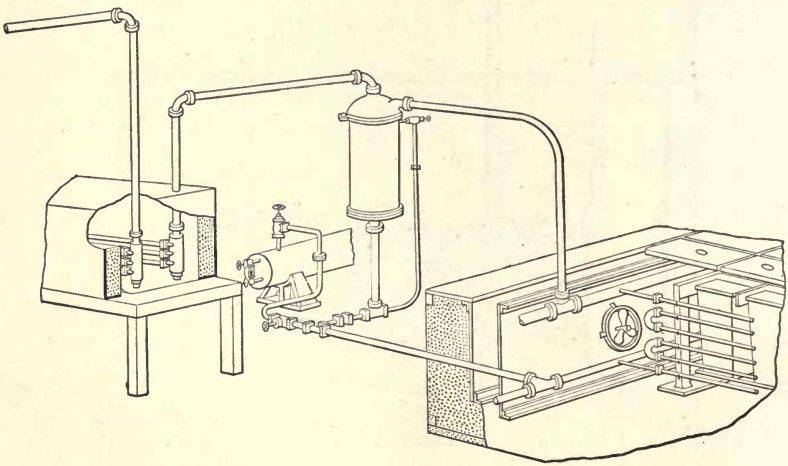


FIG. 153.—Frick Flooded System.

are controlled from one branch. In the figure liquid ammonia is carried to the bottom of the coil and the vapor is taken from the top. This is the arrangement used with the new **flooded system** which has been introduced within the last few years. As shown in Fig. 153 this system consists in supplying liquid to the lower part of the expansion coil in large quantities so that the liquid will rise through a vertical branch into the accumulator to a point above the level of the top of the expansion coil. Then as the suction pressure is decreased by the action of the compressor there will be not only a further rise in the liquid through the liquid line but also a flow of vapor and liquid

through the upper pipe leading from tank to accumulator. The accumulator acts as a separator, allowing the liquid separated to flow out at the bottom, while the dry vapor flows to the compressor or to the coils of the filtered water-storage tank. The level of liquid in the accumulator is carried near the bottom and the check valve leading to the liquid line allows liquid to flow from the accumulator.

The reason for the employment of this system is the fact that when all the pipes of the coils have liquid ammonia in them they are all effective and moreover there is a slightly better transfer of heat due to a high coefficient for liquid to liquid. In the ordinary system certain of the upper or lower pipes are filled with vapor because of the danger of getting liquid into the suction line and as a result these pipes are of little value since, after the liquid is vaporized, there can be little if any abstraction of heat, because the brine is practically at the same temperature as the vapor. Little heat can be abstracted to superheat the vapor. There is a small amount of heat transfer in these pipes, but small because the ammonia could not take up the heat rather than that the coefficient is small. In the **flooded system** (invented by J. Krebs in 1890) liquid ammonia finds its way to the highest coils and although mixed with much vapor it may remove heat from the last foot of pipe. As a result of this the pipe surface necessary for a given tonnage may be decreased from 300 lin.ft. of $1\frac{1}{4}$-in. pipe per ton to 180 feet per ton, although with longer pipe, not absolutely necessary, the efficiency of the plant is higher.

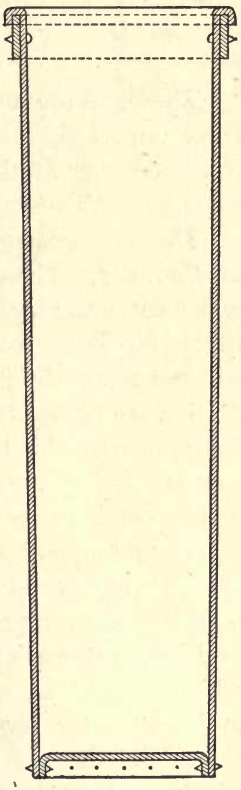


FIG. 154.—Ice Can.

The structure of **ice cans** is shown in Fig. 154. The various

manufacturers make these cans in about the same shape and size. The following sizes are used by the largest manufacturers:

ICE CAN DIMENSIONS

Weight of Ice.	Width and Breadth at Top.	Width and Breadth at Bottom.	Length Inside Over All.		Galvanized Iron, Thickness.
50	8 × 8	$7\frac{1}{2}$ × $7\frac{1}{2}$	31	32	No. 16 U.S.S.
100	8 × 16	$7\frac{1}{4}$ × $15\frac{1}{4}$	31	32	16
200	$11\frac{1}{2}$ × $22\frac{1}{2}$	$10\frac{1}{2}$ × $21\frac{1}{2}$	31	32	16
300	$11\frac{1}{2}$ × $22\frac{1}{2}$	$10\frac{1}{2}$ × $21\frac{1}{2}$	44	45	16
400	$11\frac{1}{2}$ × $22\frac{1}{2}$	$10\frac{1}{2}$ × $21\frac{1}{2}$	57	58	14

The band around the top is made of $\frac{1}{4}$ by 2-in. iron for the three largest sizes and $\frac{1}{4}$ by $1\frac{1}{4}$ for the other sizes. The iron is turned over at the top and bottom and is well riveted and soldered. All metal is galvanized.

The cans are handled by **electric hoists** or **air hoists** as shown in Fig. 155. These are mounted on light traveling cranes which carry the tanks to the ice dump. The air hose or electric cables are hung from roller hangers so placed that the loops will not reach the floor.

The **ice dump**, Fig. 156, is arranged so that when a can with ice is placed within it, the center of gravity is above the point of support and it may be easily placed in an inclined position. This motion turns on the water sprinkler. When the water has melted enough ice to free the cake, it slides out and leaves the can and frame in such a condition that the center of gravity is below the support and the frame returns to the vertical position, automatically shutting off the water. The water which issued from small holes in the pipe is caught in the tray and sent to the sewer and the ice slides free of all contamination. This is necessary, as the brine washed from the outside of the can should not come in contact with the ice.

The latest improvement in the can system of ice making has been the use of **raw water for can ice**. There have been numerous methods suggested for the production of clear ice from raw water. The **raw water can ice** of the earlier day was quite opaque and although as valuable for cooling as clear

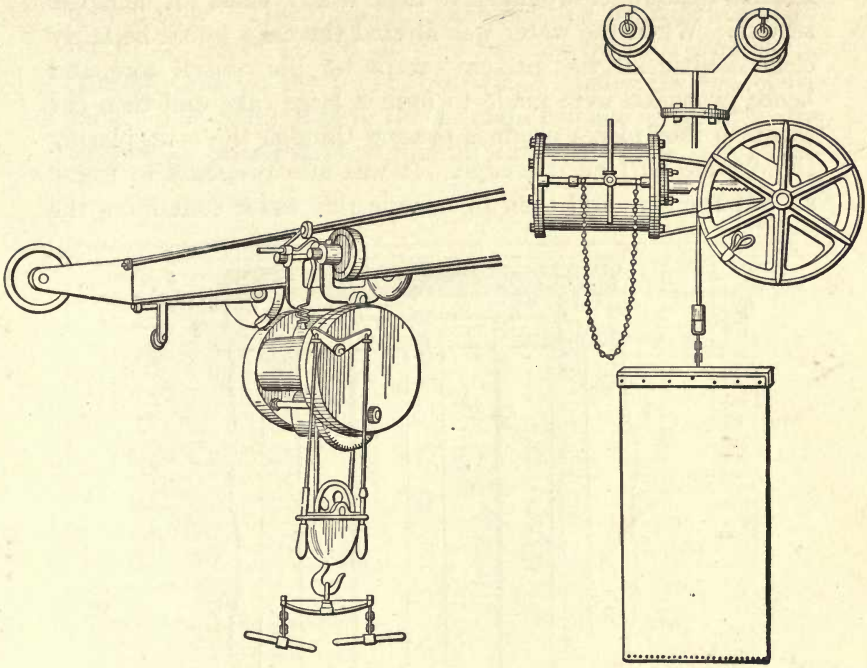


FIG. 155.—Can Hoists.

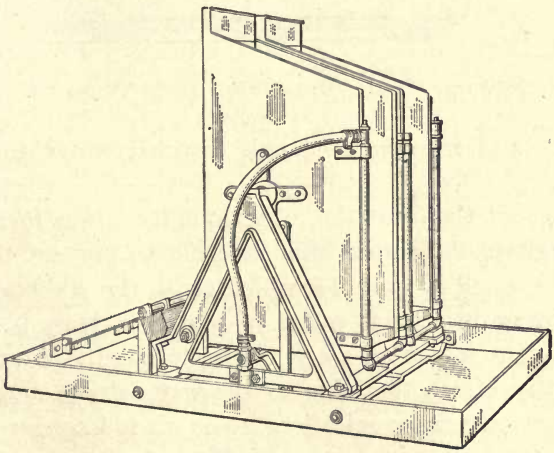


FIG. 156.—Frick Can Thawing Dump.

ice this opaque ice would have little if any value for domestic service. When the water was filtered the cake would be fairly clear if little air was present except for the central core, and hence proposals were made to form a large cake and then cut this into four pieces on lines passing through the core, placing the opaque part on one edge. It was also proposed to freeze all but the core and then to remove this water containing the

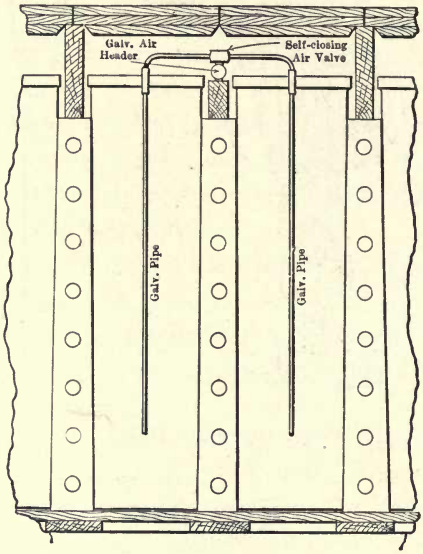


FIG. 157.—Double Drop Tube. De la Vergne Co.

impurities and introduce enough distilled water to fill this space.

To prevent the formation of opaque ice it was found necessary to **agitate the water** while freezing to prevent the retention of the small air bubbles which cause the whiteness. One method of agitating this water (patented in Italy in 1877 by Turretini) is to introduce air at the bottom of the can at a pressure slightly above that due to the water depth and this air jet produces necessary agitation. The air is introduced at the center of the bottom as shown in Fig. 160 entering in one of several patented ways. In Fig. 157 the air is introduced by

a drop tube. The tubes are connected to a tee which is placed on an air outlet of the air pipe running between every other pair of cans on the framework which carries the top of the tank. The connection to the air main is made so that the drop tubes may be withdrawn from an automatic self-closing valve,

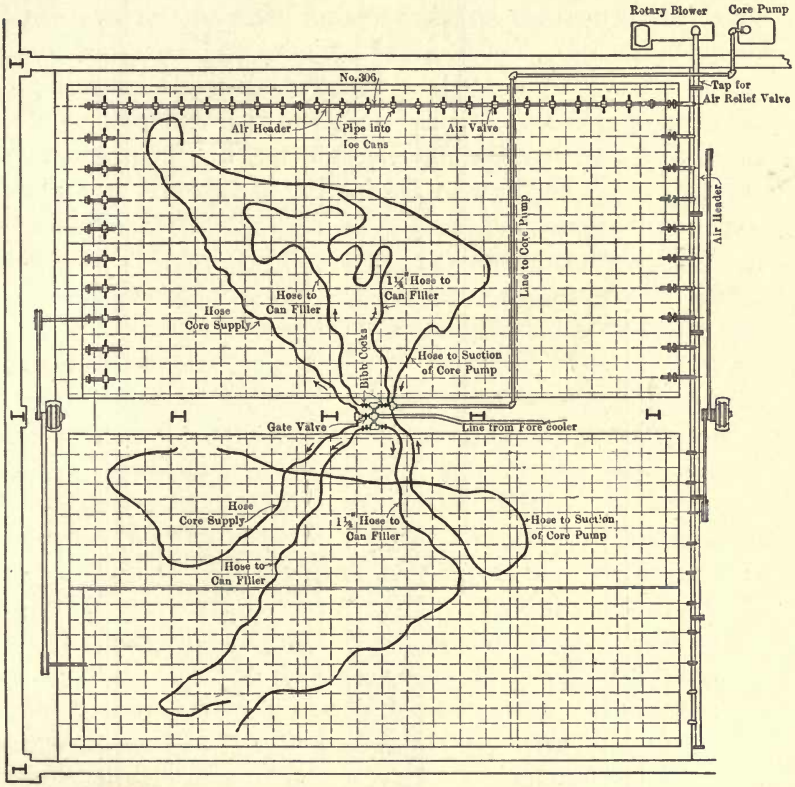


FIG. 158.—Plan of De la Vergne Raw Water Ice Plant.

which prevents air discharge when the connection is removed. As the impurities gradually collect at the center in the core space they are finally drawn out by the core pumps and the space filled with distilled water or good clear raw water which has been filtered as carefully as the original water used.

The **general arrangement of the core-pump hose and refilling hose** is shown in Fig. 158. In this figure will be seen

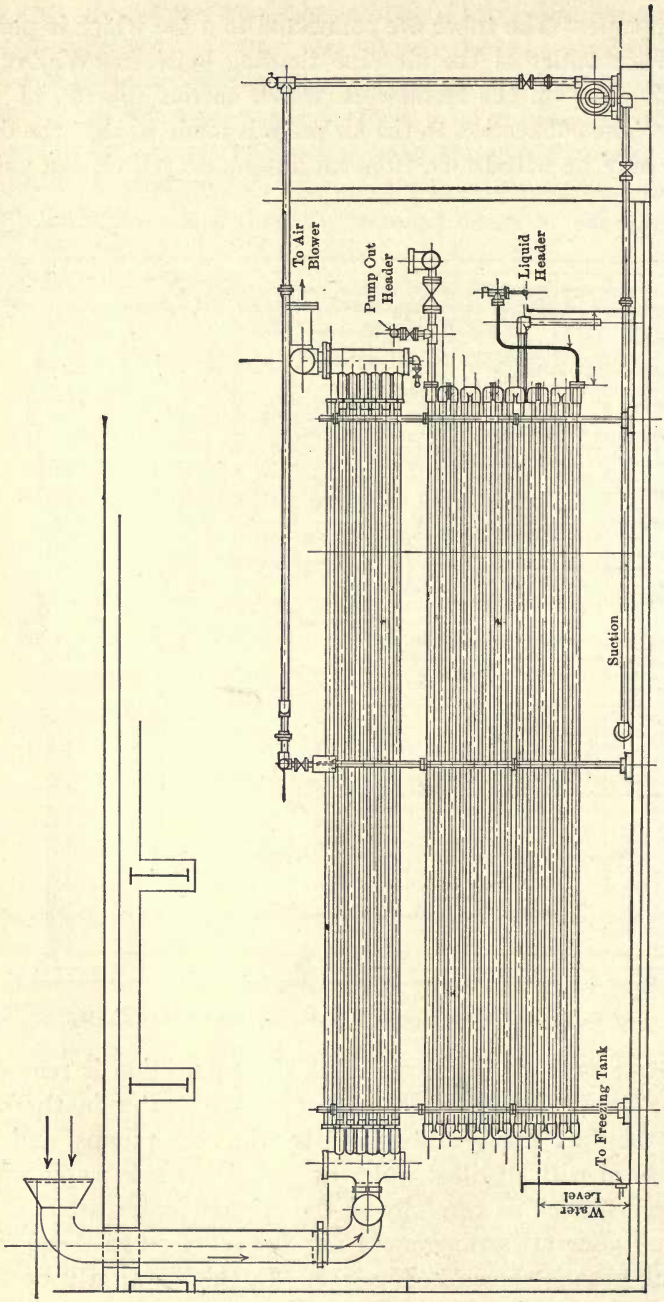


FIG. 159.—De la Vergne Air and Water Fore Cooler.

the general arrangement of one **air-supply header** running from the main header which receives air from the **rotary blower.** The core pump is connected to two lines of hose which may be put in any can for the removal of the core water. This amounts to about 10 lbs. in 300-lb. cans. The water from the fore cooler is attached to four large hose lines for filling tanks and to two small ones for filling the cores, the same water being used for each purpose. The motors driving four agitators are shown in the figure.

The **air taken into the blower** is first cooled to remove the moisture from it so that this moisture will not freeze in the air headers or drop tubes. The air is taken from the atmosphere at a high point in the plant and passed to a header from which it enters a series of pipes, Fig. 159, over which water near 32° F. is passed. In this way the air is cooled and the moisture in the air is condensed and dripped from the collecting header before the air enters the air main. The air may be taken through a screen to remove dust or any form of air cleaner may be used. The water is taken by a small centrifugal or rotary pump from the cold-water storage tank and distributed over the air coil, after which it falls over an ammonia-expansion coil, which cools it to about 32° F. This water is then collected in the tank from which it is pumped to the cans as needed or back over the air coil. The air thus dried will not clog the drop tubes as it enters and meets the cold walls of the tube. The **pressure necessary** for 300-lb. cans is about 2 lbs. per square inch, due primarily to the amount of submergence of the end of the drop tube. This air keeps the water agitated and thus wipes off any bubbles of air which might cling to the surface of the ice.

In Fig. 160 the tube *A* is fastened to the side of the can and in this the air is forced from the end *B* through the ice as it forms, leaving a small seam through which air flows, driving the water out of it and keeping the water above agitated and driving the impurities gradually to the top *D* of the can. This leaves the hollowed top of ice with all impurities above the grid *F* in the unfrozen water. This water is thrown away when

the can is taken out and the grid with the ice on it is removed, leaving clear ice of uniform length for storage. The grid also serves to freeze the center cup at the end due to the conduction of the iron. In this system the pressure has to be increased as the ice forms. This of course is automatic through the use of a valve which throttles the air discharge until resistance is brought

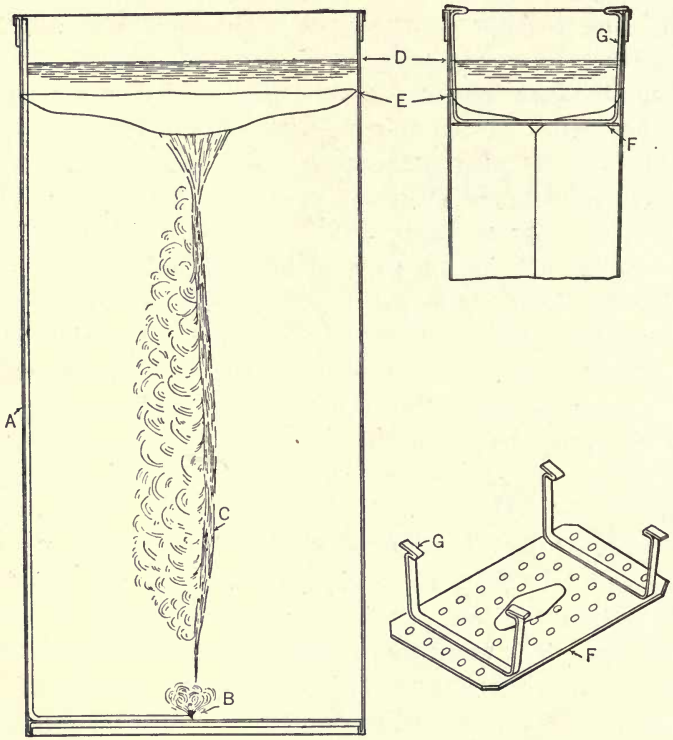


FIG. 160.—York Raw Water Can.

on by the formation of ice. The pressure at the pump remains constant, being used up in pipe and valve friction when first applied, and finally the ice friction requires the full pressure as the ice closes in. The gauge pressure is about 18 lb. per square inch. The power used in the compressor is about 0.4 H.P. per ton of capacity. The air used amounts to 1.8 cu.ft. per can per minute at start to 0.3 cu.ft. per minute after freezing.

The two methods described are those used by two large manufacturers. There are many other methods employed which have given satisfaction. For instance, the agitation has been accomplished by drawing water from the core and then allowing it to discharge back again, this back and forth motion preventing the formation of opaque ice. One of the first methods was to rock the cans and after this, agitation was by rods and then by air discharge. Another method patented by Ott Jewell is an ice can in which the brine is passed through a double wall of the can and air is introduced at bottom to agitate. The Beal patent of 1913 is similar to the York method described above. The Ulrich patent brings the air pipe in on the outside of the can.

The plate system accomplishes the result of the raw-water can system without the complication of air-pump or rocking apparatus. On account of the increased space and an increase of about 30 to 70% in the initial cost, due to the larger building, the plate system is not installed as frequently as the can system. In 1915 the De La Vergne Company stated that there were 150 plate plants in operation in the United States. Of course with the plate system almost any kind of water may be used.

The generation of the liquid ammonia may be accomplished by an absorption system as well as the compression methods described. The arrangement of the ice apparatus is practically the same when the absorption system is used to compress the ammonia. If an absorption system is placed where compressors are shown in the previous figures the apparatus would be that used by the makers of that type of apparatus for ice making.

There are more than one thousand plants in the United States, of which about 81% are operated by compression and 18% by absorption. The output of these is over twenty-three million tons of ice. The natural ice crop is probably equal to or greater than this. The curve of delivery of ice will follow the curve of temperature difference above 32° F., although there may be some changes due to manufacturing

or other application of ice. In any case it is well to draw a consumption curve to be expected or known from previous records. From this curve the capacity of the plant and the size of the storage room may be found.

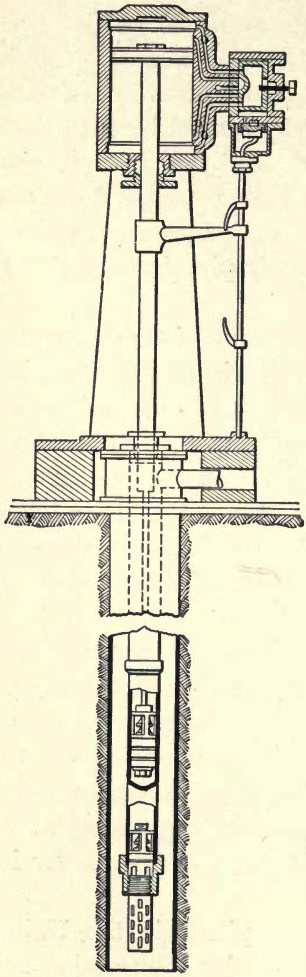


FIG. 161.—Deep-well Pump.

These various methods are applicable in special cases. The plate system is advisable with expensive fuel and very poor water, while with better water the raw-water can system may be used. When fuel is cheap the distilled can system may be used and when there is exhaust steam from other machines the absorption system should be employed. The absorption system may be operated in an isolated plant with economies as good as the compression system.

The water for ice plants is usually taken from deep wells. The wells are rarely artesian and the water has to be pumped. The pumps are of various forms. Deep-well pumps have a pump bucket operated at the end of a long rod by a steam piston in engine room, Fig. 161. An air-lift pump is one, Fig. 162, in which compressed air is allowed to enter the water pipe and by aerating the column of air and making it lighter than the water outside of the casing it drives the water out of the discharge main. Other pumps are used. The air-lift pump has the advantage of being simple to install, and of having all of the working parts accessible as well as being able to deliver a large quantity from a given well.

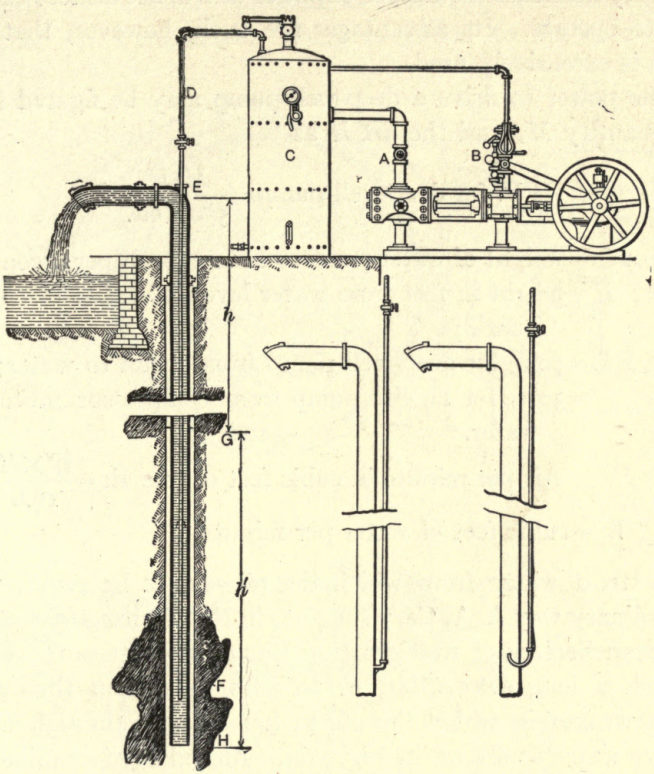


FIG. 162.—Air-lift Pump.

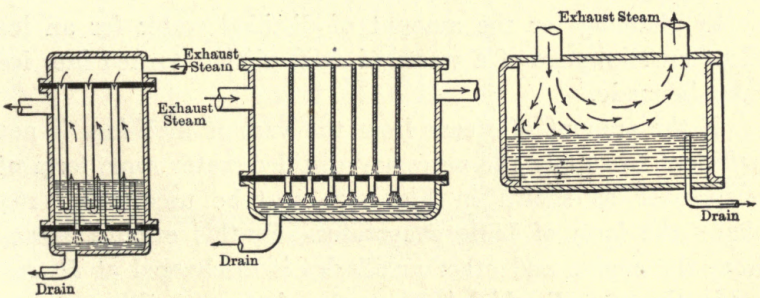


FIG. 163.—Oil Separators.

It is very inefficient in the use of power and hence is more expensive to operate. Its advantages are such, however, that the pump is extensively used.

The power to drive a deep-well pump may be figured from the quantity M_w and the lift H as

$$\text{H.P. for deep-well pump} = \frac{M_w H}{550 \times \text{eff.}}.$$

M_w = weight of water per second in pounds per second;
H = height in feet from water level in well to discharge nozzle;
Eff. = 70% for deep-well pump from steam to water;
= 30% for air-lift pump from compressor motor to water.

$$\text{Air per minute in cubic feet of free air} = \frac{W \times H}{16.8}.$$

W = cubic feet of water per minute.

To treat water from which the oil cannot be removed in the ordinary way A. A. Cary suggests in the Transactions of the American Society of Refrigerating Engineers to pass the water through a long **coke filter**, or to pass the steam through a **steam washer** in which the steam has to pass through water or over water as shown in Fig. 163. An enlarged chamber on the steam main to cut down the velocity of the steam and permit the steam to come in contact with a series of screens was also suggested.

In planning for the amount of distilled water for an ice plant an allowance of a waste of 25% of that turned into ice must be made.

If the amount of steam from the various machines is not sufficient to supply the necessary distilled water some form of evaporator as shown in Fig. 147 must be used. Fig. 164 shows the form of **Lillie evaporator**. In this exhaust steam from the engine and other auxiliaries is discharged at H and enters the tubes E, which have a reduced pressure within, due to the suction through the small holes drilled in the caps of the left-hand end of each of them from the low pressure which

exists in *B*. This low pressure is caused by a vacuum pump attached to a condenser which is joined to the evaporator

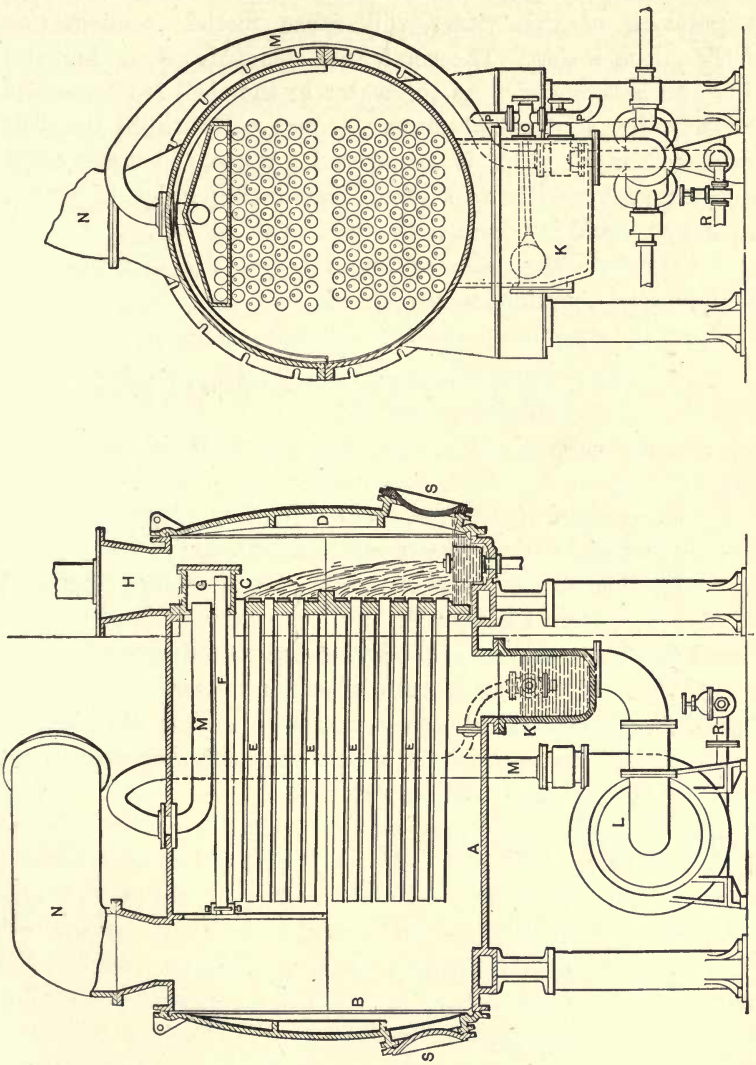


FIG. 164.—Lillie Evaporator.

by the pipe *N*. Water from the sump *K* is pumped by the centrifugal pump *L* through *M* to the *G* box whence it discharges through eleven pipes *F* and falls over the pipes *E*.

This water is heated and condenses some of the steam inside of the tube E, and the pressure is so reduced that the water will boil at a lower temperature than that of the steam inside. The evaporation of this water will cause further condensation of the steam inside. The condensed steam drops to the front of C and is removed. As the water in the shell is evaporated the sump box K does not contain enough water to lift the float and hence more water is introduced. When the water in A becomes heavy with salts after evaporating a lot of water, the heavy liquid is removed by opening R.

To compute the necessary surface and size of evaporator the following equations may be used:

From the equation

$$M_s(i'_t - i'_0) = M_w(i''_t - i''_0) + Q_e, \quad . \quad . \quad . \quad (1)$$

the amount of water M_w from the steam M_s is found. In this

M_s = weight of steam condensed per hour;

M_w = weight of water evaporated per hour;

Q_e = heat radiated from covered evaporator, computed from Chapter V;

i'_t and i'_0 = heat contents of dry steam entering and water leaving at pressure of exhaust steam;

i''_0 and i''_t = heat contents of dry steam leaving at pressure assumed in B (20° less than entering steam) and of the entering water.

The surface is figured by

$$F = \frac{M_w(i'_t - i'_0)}{K(t_0 - t'_0)}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

t_0 = temperature of entering steam;

t'_0 = temperature of steam leaving;

$K = 400$.

An important consideration in planning an ice plant is the sanitary condition around the plant. Since ice is to be used for domestic purposes and may be introduced into food or

drink, it is necessary that cleanliness be insisted upon. The men walk over the covers of the cans, and dirt on the footwear may fall into the cans, hence there should be no chance for the workmen employed in harvesting from going through places when contamination may occur. The floor of water-closets for instance should be kept so clean that nothing can cling to the footwear. The water-closets should not be placed so that workmen would have to pass through stable yards or over roadways to reach them. In many ice plants conditions exist such that men must pass through regions where the boots take up this contaminating matter. The condition of the water-closet should be bright and clean. Money spent here on tile work and terazzo flooring is not wasted.

The wells or springs from which the water is taken must be placed so that they may not be contaminated by materials blown by the winds or from ground or subsurface seepage. The use of cesspools for water-closets should not be tolerated, especially where wells or springs are used for water supply.

To keep the plants clean they must be so constructed that dirt will show, and hence the covers, walls and all parts of the plant should be painted white. It is also advisable to have no one walk over the tank tops who does not put on rubber over-shoes which are not worn anywhere else.

Freezing Tanks. Freezing tanks are usually made of steel plate with insulation beneath and around them. This is shown in Fig. 141. The insulation is sufficiently heavy to cut down the heat loss to a low value. The thickness is fixed by finding the minimum yearly cost due to heat loss, interest, depreciation, taxes, insurance and maintenance. The tanks for the plate system are sometimes made of timber. These must be carefully braced whether of wood or metal because of the depth. Reinforced concrete has been used for brine tanks. The bottom and sides are made of 6 or 8 ins. of concrete with $\frac{5}{8}$ in. reinforcing bars placed 1 ft. apart and 2 ins. within the concrete from the brine side. This is followed by 2 ins. of cork board and then five layers of tar felt put on with hot coal tar, on top of which is placed 2 ins. of concrete with a smooth finish. To cut

down the heat transfer a thick layer of cinder may be used around the tank, 16 ins. of dry cinder being equal to 2 ins. of cork. There is a difference of opinion as to the advisability of using concrete for the brine tank, as some claim calcium chloride disintegrates the concrete, but others say that concrete is perfectly satisfactory.

In planning the size of brine tanks the **time of freezing** must be assumed and sufficient cans or plate tanks must be installed to give the required capacity. In plate work one freezing tank should be large enough to give the required output. If the weight of ice is taken $57\frac{1}{2}$ lbs. per cubic foot, the volume of the plates for the given tonnage, assuming a thickness of 12 or 14 ins., may be found, and from this the volume of the tank, the

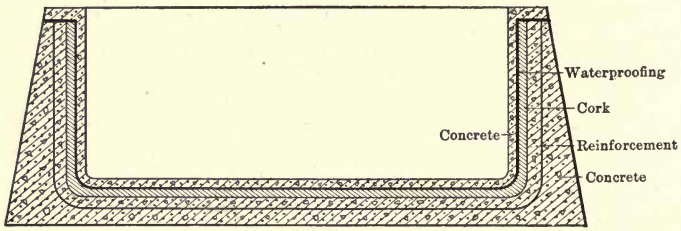


FIG. 165.—Reinforced Concrete Brine Tank.

depths being about 10 ft. and the widths 16 ft. The time of freezing the plate ice is given by Macintire as

$$h = \frac{21a^2}{32 - t_f}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

h = hours of freezing;
a = thickness in inches;
t_f = temperature in refrigerating pipes in deg. F.

This takes about six days, so that six or seven tanks are used. The number of cans is found in the same way. If the smaller thickness of the can is represented by a, then the time is given by Macintire as

$$h = \frac{4.2a^2}{32 - t_f}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

a = thickness of can at top in inches.

In this way the total output during the freezing is

$$\frac{h \times \text{tons}}{24}.$$

And the number of cans is given as

$$\frac{h \times \text{tons}}{24 \times \text{Wt. per can}} = \text{number of cans.} \quad . \quad . \quad . \quad (5)$$

Ordinarily fourteen 300-lb. cans are used per ton of capacity. This means fifty hours for the formation of the ice. If the number of cans is increased it means that there is a longer time for the ice formation and hence a smaller difference in brine and water temperatures, which means a higher back pressure and less work, while a decrease in the number means a lower brine temperature to give the smaller time for freezing. This means a lower back pressure on the compressor and hence more work. It is well to compute the yearly cost of ground, building and equipment against the cost of power in figuring the number of cans. To compute the temperature of the brine a value of 2.6 for K is used. This same thing is true in regard to the number of tanks in the plate system. To show this Thomas Shipley has computed the following table:

EFFECT VARIATIONS IN CAN ALLOWANCE HAVE ON HORSE-POWER REQUIRED TO PRODUCE ONE TON OF ICE. (With Single-acting Compr.)

1	2	3	4	5	6
No. 300-lb. Cans Per Ton Ice Making.	Average Brine Temperatures Needed to Produce Ice. °F.	Rate of Heat Transmission B.T.U. per Sq.ft. per Hr. 1° MD. for Pipes.	Temperature Required in Pipe. °F.	Corresponding Evaporating G. Pressure. Lbs.	Total Brake H.P. per Ton Ice Making 185 Lbs. C. P.
10	7	15	−3.3	13.3	2.77
12	11	15	+0.7	16.2	2.56
14	14	15	3.7	18.5	2.45
16	16	15	5.7	20	2.352
18	18	15	7.7	21.7	2.27

Note.—Evaporating surface in the freezing tank assumed in this table is 108 sq.ft. or 250 ft. of 1¼-in. pipe per ton of ice.

Note.—Tables are based upon the water to be frozen being delivered to the cooling and freezing system at 70° F.

Work done by cooling system = 30 B.t.u. per pound of ice.

Work done by freezing system = 200 B.t.u. per pound of ice.

Expansion Coils. In the coils of brine tanks the liquid ammonia may enter the upper or lower pipe. When the flooded system is used the liquid is introduced at the bottom, and it seems unreasonable to bring it in at any other point if the coil is to receive its full supply of liquid. In this case the vapor header may be drained to the low liquid line to return any liquid unevaporated. This really gives a flooded system. Before the wide introduction of the flooded system there was a great difference of opinion over this point among the refrigerating engineers, but this matter seems to be settled by the adoption of the flooded system.

In some cases the brine may be cooled in a brine cooler on one of the types shown in Figs. 19, 81 and 82 and the brine pumped to the freezing tank. Such a device is not so good as the use of expansion coils in the freezing tank, as this method keeps the brine at a low temperature throughout by abstracting heat from it as it abstracts heat from the freezing water.

The amount of this surface in a brine tank is figured by allowing a value of K of 15 due to the low velocity of the brine over the coils. In a brine cooler especially of the double pipe type or the shell type a much higher value of K is used, due to the higher velocity of the brine. Since this method is rarely used the case of the expansion coil in the tank will be considered. It has been found that 120 to 150 lin.ft. of $1\frac{1}{4}$-in. pipe is sufficient to care for a ton capacity with ordinary coils and about 80 ft. have been found necessary in the flooded system. In any case it is a matter of abstracting the heat and if the surface is cut down the back pressure must be decreased to give the necessary temperature difference. This means greater power for the same refrigeration. One must compute the yearly cost on investment on pipe against the cost of power. Assuming sixteen cans per ton, Thomas Shipley has computed a table showing the effect of change of pipe length.

The powers required are the powers applied to the compressors to drive them, whether by belt, direct-connected motor or engine. The auxiliaries require about 0.3 to 0.4 H.P.

EFFECT OF VARIATIONS IN EVAPORATING SURFACE ON HORSE-POWER REQUIRED TO PRODUCE ONE TON OF ICE WITH SINGLE-ACTING COMPRESSOR.

1	2	3	4	5	6	7
Lineal Ft. of 1¼-in. Pipe per Ton of Ice Making.	Sq.ft. Pipe Surface, External.	Rate of Heat Transfer for Pipe = K.	Average Temperature of Brine. °F.	Temperature in Pipe. °F.	Gauge Press in Coil. Lbs.	Total Brake H.P. per Ton of Ice Making at 185 Lbs. Comp. Pres.
150	65	15	16	−1.1	14.85	2.661
200	87	15	16	3.2	18.1	2.468
250	108	15	16	5.7	20.0	2.352
300	130	15	16	7.45	21.5	2.279
350	152	15	16	8.7	22.5	2.218

The pressure of compression has an important bearing on the efficiency of the plant. An endeavor should be made to use as cool water as possible for the condenser to keep this pressure low.* The effect of this is seen in the following table by Thomas Shipley:

POWER REQUIRED PER TON OF ICE AT DIFFERENT CONDENSING PRESSURES WITH A SINGLE-ACTING COMPRESSOR WORKING UNDER A SUCTION PRESSURE OF 20 LBS.

Condensing pressure	85	105	125	145	165	185	205	225	245
Corresponding Tem.	47.6	58.6	68.1	76.6	84.2	91.0	97.3	103.2	108.9
H.P. per ton of ice..	1.162	1.408	1.646	1.874	2.114	2.352	2.587	2.851	3.098

The rule for the number of tanks and size of the expansion coil has been worked out on the assumption of 200 B.t.u. per pound of ice in the freezing tank and 30 B.t.u. per pound in the cooling tank. Of course there are average results, but in some cases the quantity must be computed owing to peculiar conditions. The **heat to be removed** in the cooling system per pound of ice made, if 15% excess is allowed and water is at $t°$ F. and is to be cooled to 32 is

$$Q_c = 1.15q' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

*If water has to be pumped and is bought, a calculation must be made for the cost of water and of power for the compressor and for the pump for different quantities of water and the one giving the best result used.

The **heat of fusion** is 143.4 B.t.u. and the **specific heat of ice** is given by

$$c = 0.5057 + 0.001863t - \frac{0.004}{t^2}. \quad . \quad . \quad . \quad . \quad (7)$$

c = specific heat ice;
t = temperature of ice in degrees C.

This has been tabulated by Dickinson and Osborne as follows:

HEAT TO FREEZE ONE LB. OF WATER AT 32° AND TO COOL ICE TO TEMPERATURE.

t	Q	t	Q	t	Q
−20	167.2	−2	159.5	16	151.2
−18	166.3	0	158.6	18	150.3
−16	165.5	2	157.7	20	149.3
−14	164.7	4	156.8	22	148.4
−12	163.8	6	155.9	24	147.4
−10	163.0	8	155.0	26	146.4
− 8	162.1	10	154.1	28	145.4
− 6	161.3	12	153.1	30	144.4
− 4	160.4	14	152.2	32	143.4

The **heat used then will be**

$$1.15Q_f = Q = 1.15 \times 160 = 184 \text{ B.t.u. approx.} \quad . \quad . \quad (8)$$

Although only 184 B.t.u. are needed, the radiation will be an additional amount, making 200 B.t.u., used by Shipley. The radiation loss may be computed.

The **size of pipes** for brine, ammonia and water are computed by method of Chapter VI.

From the **curve of ice consumption**, as shown in Fig. 166, the main demand for ice can be found and the question arises: Is it better to put in enough ice-making capacity to carry the peak load, having idle machinery during a large part of the year, than to install apparatus for the mean capacity and operate at this capacity, storing at time of low demand to carry the amount demanded at times above the mean curve? This problem is one which can be computed. The extra apparatus

and plant needed beyond that for mean load is found and the cost of interest, depreciation, taxes and insurance is compared with the cost of storing, loading and removing ice, including the cost of the building.

To store ice and hold it from spring to midsummer and then take it from storage costs 25 to 30 cents per ton according to W. E. Parsons. J. N. Briggs increases this by the yearly cost of the storehouse, 15 cents brings the cost to 40 or 45 cents per ton. This includes the expense of holding, and in this

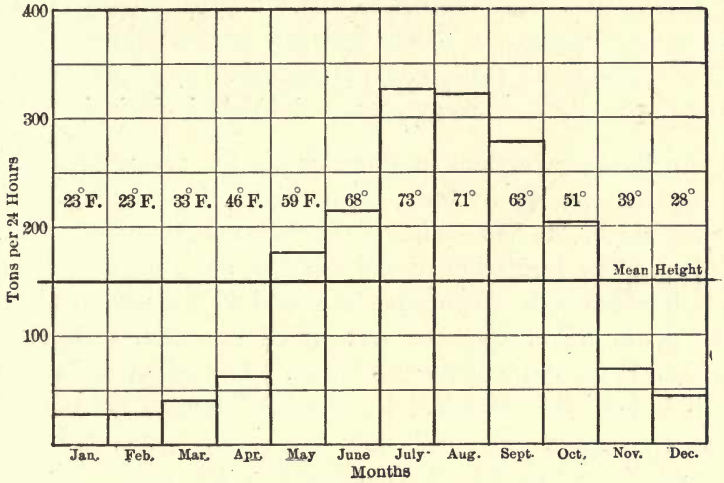


FIG 166.—Curves of Tons per Day. Arranged for months for Troy, N. Y. Monthly average temperature for forty years.

problem the cost of insulation is compared with that of absorbing the heat transferred to determine the amount of insulation. After this is fixed the amount of heat loss is found and cared for by melting ice or by refrigerating coils. The latter method is the better, as it is cheaper to cool the room than to make an equivalent amount of ice to do this. The data from a storehouse of this kind will be mentioned.

From the amount of heat to be supplied by the brine coils in the freezing the surface may be computed by assuming the temperatures of the water in cans, brine, and ammonia. Call these t_w, t_b, and t_a,

Can surface in sq.ft. per ton of ice

$$=\frac{2000\times[1.15Q_f+Q_r]}{24\times 2.6[t_w-t_b]}=\frac{6410}{32-t_a}. \quad . \quad . \quad . \quad . \quad (9)$$

Coil surface in sq.ft. per ton

$$=\frac{2000\times[1.15Q_f+Q_r]}{24\times 15[t_b-t_a]}=\frac{1111}{t_b-t_a}. \quad . \quad . \quad . \quad . \quad (10)$$

The **amount of refrigeration** for plant per ton of ice

$$=\frac{[Q_c+1.15Q_f+Q_r]2000}{199.2\times 24} \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

$$=\frac{Q_c+1.15Q_f+Q_r}{2.39}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

An **ice-storage plant** in Philadelphia for 10,000 tons of ice is 113 ft. long, 78 ft. wide, 60 ft. high. It is built of brick 22 ins. thick. It has a 6-in. reinforced concrete roof carried on girders. The insulation of the walls is two 2-in. cork boards held in place with cement and furnished with a cement plaster. The floors are of concrete over which two 2-in. cork boards are laid in hot asphalt for insulation. The ceiling is insulated with 3 ins. cork attached by Portland cement to the reinforced concrete. To refrigerate the room, 15,000 ft. of 2-in. wrought-iron pipe for direct expansion are used. This held the room at 22° F. in the warmest weather. To handle the ice, plunger elevators with oil and brine as the working fluid are used.

Ice is stored in these houses in such a way that the ice will not press against the walls and the blocks should be so placed that they will make a stable pile. This is important.

In **distributing ice** it is well to use a map of the city and plan routes so that they will not overlap. Inspectors must be employed to watch men and foremen to direct their work. To guard against loss of money ice books are sold by drivers, who are held responsible for them, and in addition the amount of ice delivered to the driver and the amount sold by him must receive a daily check. It is best to use a single-horse wagon

in charge of one man, since where two men are on the wagon there is apt to be talking, drinking and more waste of time. To encourage better work it is well to give a bonus.

The use of 3-ton automobile trucks has been found to cut down the expenses of distribution in the saving of time in reaching the point of delivery. This is especially true if the truck is used to carry ice to the delivery wagons. In using automobile trucks the work must be arranged to handle ice quickly. There must be no waiting, as the fixed charges are so great that unless a large volume of work is done, there is loss.

Solicitation of trade in an unobtrusive way, care in adjusting all complaints and judicious use of advertising will bring good results. It is absolutely necessary for the foreman to be acquainted with the kind of service given to consumers.

CHAPTER VIII

OTHER APPLICATIONS OF REFRIGERATION

THE use of refrigerating in various industries is increasing. In a recent list of applications over one hundred industries were mentioned in which refrigeration played an important role. A few of these will be described.

In the **manufacture of candy**, especially chocolate-coated candy, there is a necessity for a uniform temperature to set the chocolate and to give uniform results. Cool air is blown into the rooms to keep them at a temperature of about 68° F. as chocolate cannot be dipped above 72°. This air may either enter from an overhead duct at 50° F. and be used to cool the room enough to set the chocolate, or air may be introduced into **setting boxes** shown in Fig. 167. The chocolates are placed in this box as soon as one plate has been filled with dipped chocolates. By pressing a pedal the plates in the box are raised to admit a new plate from beneath. The plates are separated by distance pieces so that there is no danger of the candies being mashed by contact. Cool air is introduced into the interior of the box from the duct *A* and before entering the room this air chills the candy, setting the outer coating. The heat which it removes would have to be removed in any case to hold the room at 65 or 68°, and so this apparatus requires no extra refrigeration, but it applies the cool air where it will do the greatest good. The springs *B B* hold the plates in position. This is employed by Harter & Co. of Ohio. They have a $6\frac{1}{2}$-ton machine, cooling a bunker $5' \times 5' \times 22'$, using five stands of 2-in. direct-expansion pipes 12 ft. high and 20 ft. long for cooling. A 52-in. Buffalo Forge Fan at 120 R.P.M. drives the air through a 24-in. riser to 16-in. pipes with 3-in. branches leading to the cold boxes. This saves refrigeration,

as the room need not be cooled to such a low temperature. In the plant under discussion the 72 boxes, each caring for 150 lbs. of chocolate per day, required $6\frac{1}{2}$ tons while it would require 15 tons for the ordinary cool room according to the designer.

At the Baker Chocolate Plant six 100-ton absorption machines are used to supply 40,000 lin.ft. of 2-in. galvanized

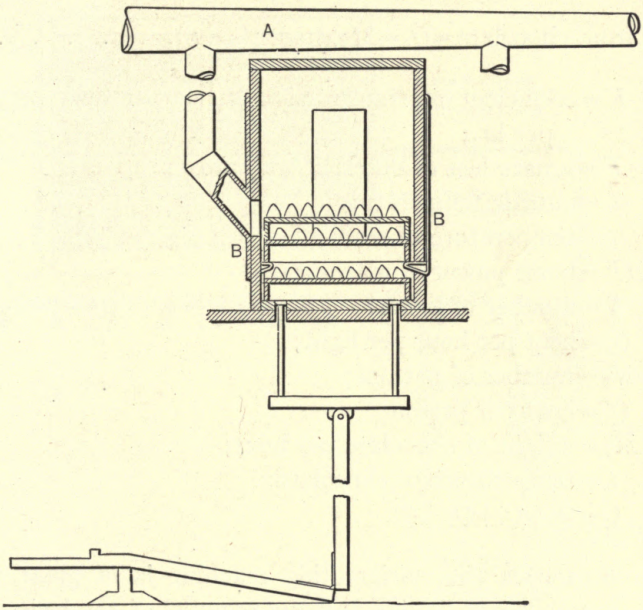


FIG. 167.—Setting Box for Chocolates.

iron pipe. The pipe is carried throughout the plant and bunkers and fans are placed where required.

The **specific heat of chocolate** is given as 0.9 by Siebel, and he also recommends the air to be supplied at such a saturation that the relative humidity at room temperature is 72%.

The heat removed in this case is similar to that for the blast furnace as given on p. 324. The heat from the walls of the buildings, machinery, lights, persons and chocolate are given by the following:

Heat leaking through walls per hour

$$= Q_e = \Sigma KF(t_a - t_r). \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Heat from machines and lights per hour

$$= Q_l = 2546 \text{ H.P.} + N_l Q_g. \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Heat from persons $= Q_p = N_2 Q'. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$

Heat from chocolate $= Q_c = M_c \times 0.9 \times (t_c - t_r). \quad . \quad . \quad . \quad . \quad . \quad (4)$

K = coefficient of transmission B.t.u. per deg. per sq.ft. per hr.;
F = square feet of different wall areas;
t_a = outside temperature;
t_r = temperature of room;
HP = horse-power of machines;
N = number of lights;
Q_g = heat per hour per light;
N_2 = number of persons;
Q' = heat per person;
M_c = weight of chocolate per hour;
t_c = temperature of chocolate;
$Q = Q_e + Q_l + Q_p + Q_c$.

In **breweries** the refrigerating machine is of great value. Here the cooling of liquids and the removal of the heat of fermentation are the chief applications.

After the beer is brewed the solution, known now as wort, has to be cooled and this is done usually in a **Baudelot cooler**. The Baudelot cooler, Fig. 168, consists of a series of horizontal pipes formed in a coil through which cold water is passed and below this is another coil in which liquid ammonia is allowed to boil or brine or cold water is circulated. Over these coils the hot wort is distributed. The wort is exposed to the air and will take up oxygen, thus throwing down certain matter in solution, and also there is some solid matter thrown out due to the cooling. The room in which this occurs is usually

enclosed in glass or copper so that it may be kept clean and free from foreign bacteria, which would produce growths not desired in the fermentation. This cooling is intended to reduce the temperature to prevent bacterial growth should any enter. The heat removed from the wort depends on the range of temperature and the specific heat. The specific heat varies from 0.941 at specific gravity 1.032 to 0.861 at specific gravity 1.0832, with a negative allowance of 0.00015 for each degree

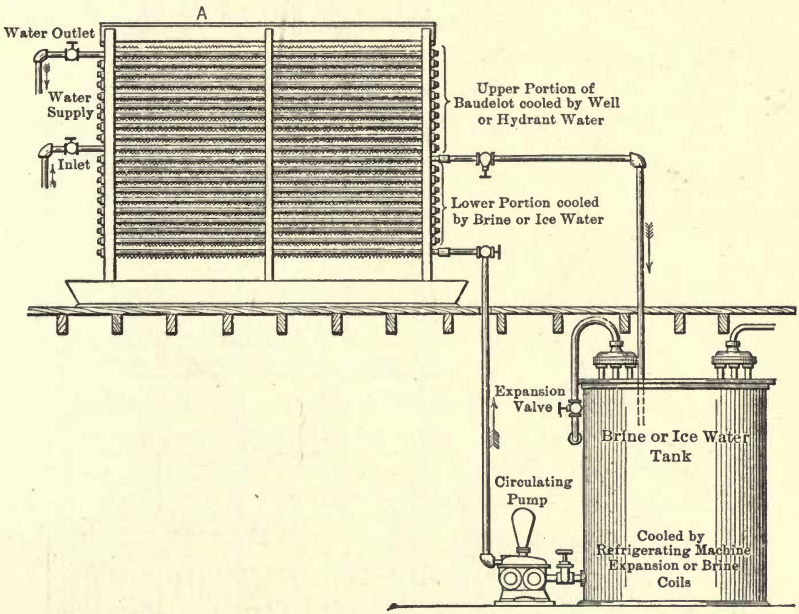

A
Water Outlet
Water Supply
Inlet
Upper Portion of Baudelot cooled by Well or Hydrant Water
Lower Portion cooled by Brine or Ice Water
Expansion Valve
Brine or Ice Water Tank
Circulating Pump
Cooled by Refrigerating Machine Expansion or Brine Coils

FIG. 168.—Frick Baudelot Cooler for Beer Wort.

above 60° F. The ordinary drop in temperature is from boiling to 110° F. in a storage vat and then the wort is passed over the cooler and reduced to 70° F. on the upper coil and to from 40 to 50° F. on the lower coil.

The beer is then taken to fermenting tubs where the sugar, formed in the operation from the starchy matter by the diastase which was produced by the change of barley to malt, is split up into CO_2 and alcohol by the action of yeast added to the wort for the fermentation process. The bacteria of the yeast

take up oxygen and also nitrogen. This action produces heat and as a rise in temperature would make brewing difficult

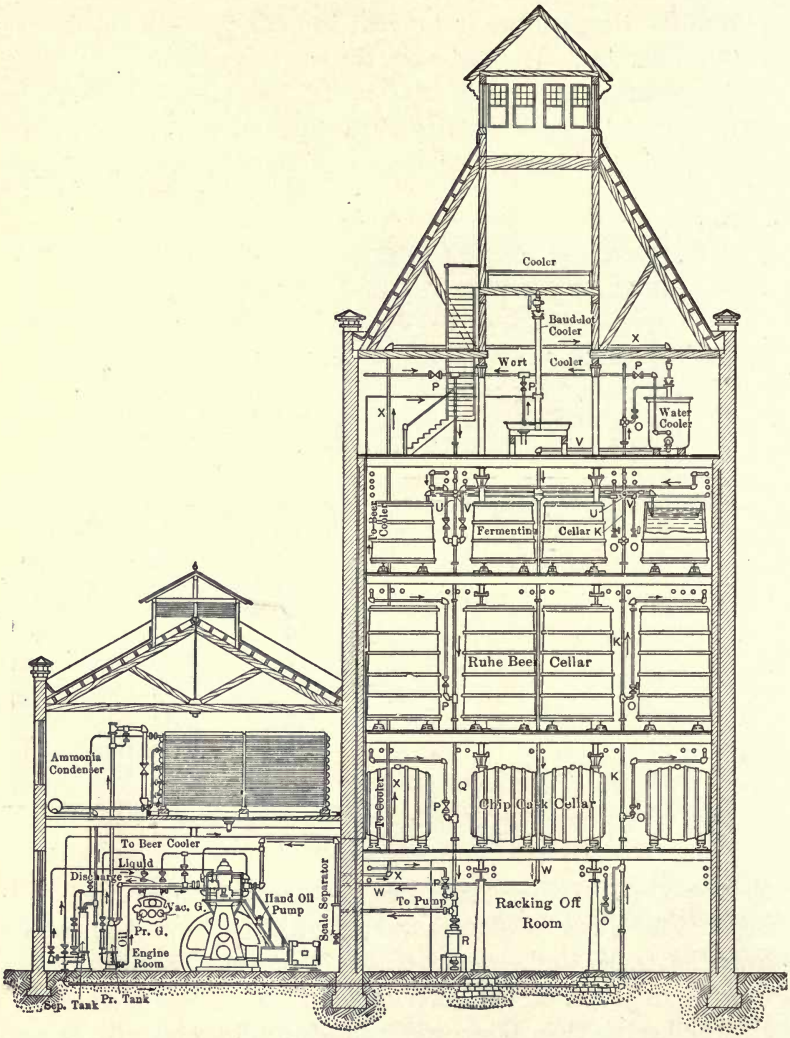


FIG. 169.—Brewery Plant Showing Cellars and Beer Cooler. After De la Vergne.

and because the beer is to be reduced gradually in temperature, it is necessary to cool this liquid in the fermentation tubs. This is done by circulating cool water through attemperating

pipes. The water is cooled in the attemperator by brine or direct expansion.

After this the beer is placed in storage tubs to age and finally put into large casks, where it is properly finished off. From this point it is placed in kegs for shipment. This is known as racking.

The **heat removed** is the heat leakage through the walls, Q_w; the heat from the Baudelot cooler, Q_b; the heat from the attemperator, Q_t, for the fermentation.

The heat Q_w is computed in the manner mentioned before as soon as the temperatures of the rooms are fixed. The fermenting room is kept at 42° F., the storage rooms at 33° F., the cask rooms at 36° F. and the racking room at 32° F.

The heat Q_b is given by

$$Q_b = \text{vol.} \times 62.5 \times \text{sp.gr.} \times c(t_b - t_f), \quad . \quad . \quad . \quad . \quad (5)$$

V = volume of wort per hour in cubic feet;
sp.gr. = specific gravity = 1.05 mean;
c = specific heat = 0.9 mean;
t_b = temperature from brewing = 150 to 190° F.;
t_f = temperature to tubs = 40° F.

According to Siebel the heat removed in fermentation is given by

$$Q_f = M_m \times 330, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

M_m = lbs. of maltose split up per hour into CO_2 and alcohol;
330 = B.t.u. produced by the breaking up of 1 lb. of maltose.

This may be written as

$$Q_f = 650 \times M_a. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

M_a = lbs. of alcohol produced per hour.

This usually amounts to a ton of refrigeration per 40 to 60 barrels per day. The **total refrigeration of the brewery** amounts, according to Siebel, to a ton for every four barrels per day.

The **amount of surface** may be computed by the methods of Chapter V. Ordinarily the Baudelot wort coolers are made of ten 2-in. pipes 16 ft. long for fifteen barrels of beer per hour to cool the wort from 70 to 40 by the use of direct ammonia expansion. With brine these pipes would care for ten to twelve barrels per hour. The water portion of the cooler to cool the wort from 170 to 150 to 70 would be made of about the same number of pipes. These pipes may be made of copper. In the attemperators, coils are made usually of a coil diameter of two-thirds the tub diameter. Twenty-four square feet is allowed by Siebel per 100 barrels of wort. The rooms for storage of hops should be held at about 36°.

The **cooling of air** is one of the modern applications of refrigeration. This cooling is not always undertaken to obtain cool air, for it is used in blast-furnace work where warm air is needed, but in this case the cooling is to reduce the moisture content of the air. Air at any temperature may contain a definite quantity of moisture, and when this amount is present it is said to be saturated. The amount of moisture per cubic foot to saturate the air is different for each temperature. It amounts to the weight of 1 cu.ft. of steam at that temperature. The air and moisture are really the mixture of several gases and a vapor, and by Dalton's law the amount of each constituent is proportional to its partial pressure. The moisture or steam may exert the pressure corresponding to its temperature if saturated, and with this pressure the weight must be that required to produce saturation. If the air is not saturated the moisture is in a superheated condition. The ratio of the amount present to the amount required to saturate the air is known as the relative humidity, as was stated on p. 50. The method of finding the relative humidity was given on p. 175. From Fig. 92 it is seen that air of relative humidity 0.90 and of temperature 85° contains 13.7 grains of moisture per cubic foot. If this air is cooled to 82° F., it will be saturated, and if cooled to 70°, it can only contain 8 grains of moisture. So that 5.7 grains must be thrown out of suspension. If the air is cooled to 36° F., the air contains only 2 grains per cubic foot.

Thus in the summer, warm air may be passed over a set of coils containing cool brine and when this air is delivered into building it will be cooled and contain less moisture than in its first atmospheric state.

If atmospheric air is always cooled to a low temperature, say 34°, before being introduced into a system, the air will always contain the same amount of moisture no matter what the original relative humidity of the warm air has been. It is this fact which has been applied by **James Gayley** to the **drying of blast-furnace air.**

In an article by Gayley in the Transactions of the American Institute of Mining Engineers, he points out that in the Pittsburg district the variation of monthly average temperature is from 31.7 to 76.2 during the year, the amount of moisture per cubic foot in these two cases being 1.83 grains and 5.60 grains. The moisture varies from 0.56 to 8.78 grains, changing by large amounts even in one day. In January this change was from 0.56 grain to 0.88 grain on the same day and from 5.55 to 5.74 grains on a day in July. This means that although the ore, and coke and limestone have a definite composition within 10% variation, the moisture content may vary 100%. This results in a varying amount of coke to care for the dissociation of the moisture and a difference in iron produced. The ordinary furnace uses about 40,000 cu.ft. of air per minute and this contains 40 gallons of moisture per hour for every grain. To make this uniform Gayley proposed to cool the air and after trying an experimental installation he applied it to the Isabella furnace at Etna, Pa.

In this plant the air is drawn over the pipes *A*, which are supplied with brine. These pipes are arranged in three coils of twenty-five 2-in. pipes, 20 ft. long. The three coils are placed above each other and are supplied from the headers *C* and discharged into the 4-in. headers *B*. The coils are arranged with staggered 2-in. pipes and there are sixty coils in the width of the bunker room, making 180 coils of twenty-five pipes. Crosswalls divide the coils into four sets. There are 90,000 lineal feet of pipe. The room is 44 by 28 by 36 ft. and is lined with 2-

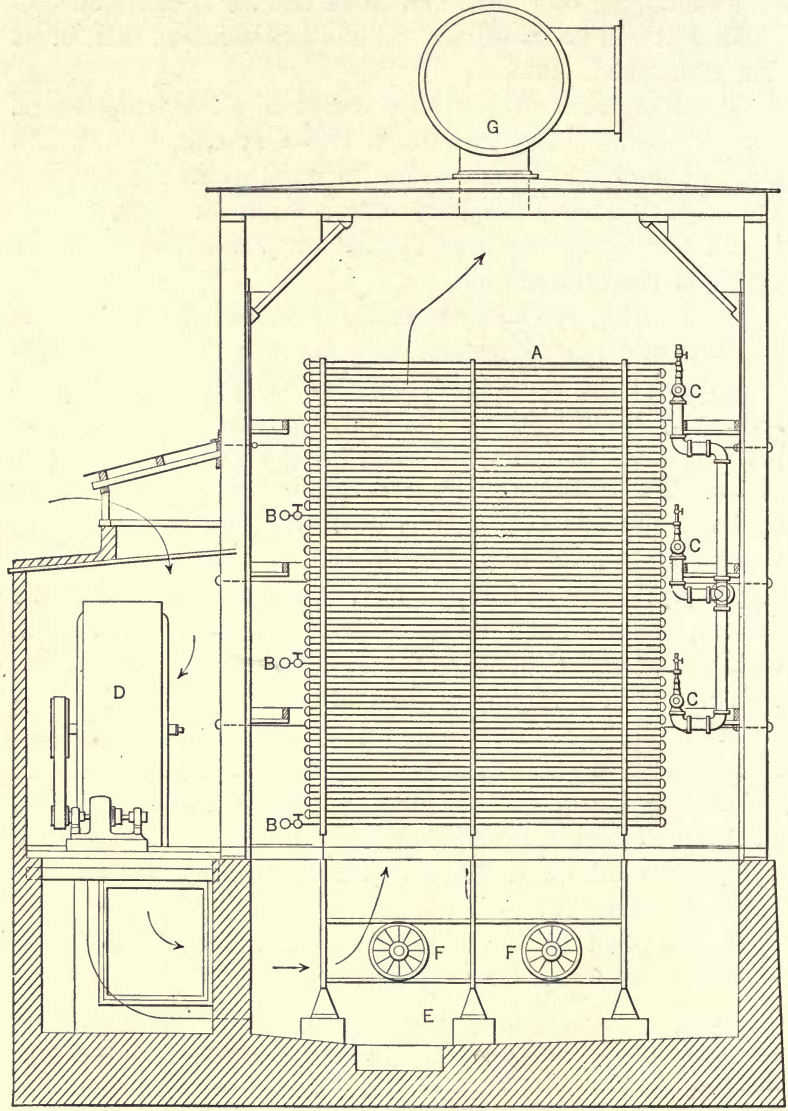


FIG. 170.—Gayley Air Cooler.

in. cork. Air is drawn in by fan *D* and put into the space *E* under 1.2 oz. pressure to care for the frosting of the pipes and the closing of the air passage. The fans *F* keep the air evenly distributed over the brine coil. The air finally enters the 6-ft. pipe *G* and passes to the blowing engine and after compression it is sent to the hot-blast stoves. The moisture taken out amounts to from 3000 to 5000 gallons in twenty-four hours. Some of this freezes on the pipe as the brine enters at 16° F. and leaves at 33°. The defrosting is necessary every fourth day, and for that reason the brine is shut off of one of the four compartments and the warm water from the ammonia condenser is passed through the pipes for two or three hours. The plant is equipped with two 225-ton compressors requiring 460 H.P. There are thirty-seven coils in the atmospheric ammonia condenser and twenty submerged double-pipe brine coolers placed in a brine tank 7 ft. 6 in. deep and 22 ft. 6 in. long. The coolers are made of twelve double pipes, 2 ins. and 3 ins. diameter and are 17 ft. 12½ in. long. The brine in the tank and in the inner tube is cooled by the ammonia in the annular space. There are 40,000 gallons of $CaCl_2$ brine of sp.gr. 1.2 in the system. The brine pump and fan *D* take 75 H.P. The total power needed is 535 H.P., while the three air compressors use 3 times 671 or 2013 H.P. in place of 3 times 900 or 2700 H.P., as was required with the warm air. The smaller power is due to the smaller volume occupied by the cooler air with small vapor pressure. There is a slight saving in power, but the main saving is in the amount of coke used and in the uniformity of operation; 358 tons of iron were made with 2147 lbs. of coke per ton originally while 447 tons per day were made with 1726 lbs. of coke per ton after the installation of the dry blast. This plant was started in August, 1904, and since then a number of plants have been installed. In general the output may be increased by 10% and the saving in coke is 15% when this apparatus is used.

Gayley has patented a scheme of passing the air through two coolers in series and using cool liquid to abstract the heat. In this case air is blown in at *A* and passes up through grids or

baffles over which a cold liquid such as water falls. The water is pumped by the centrifugal pump *C* to the top of the tower where it is discharged over brine or direct-expansion coils *D*, which cools off the water and this flows down over the grids.

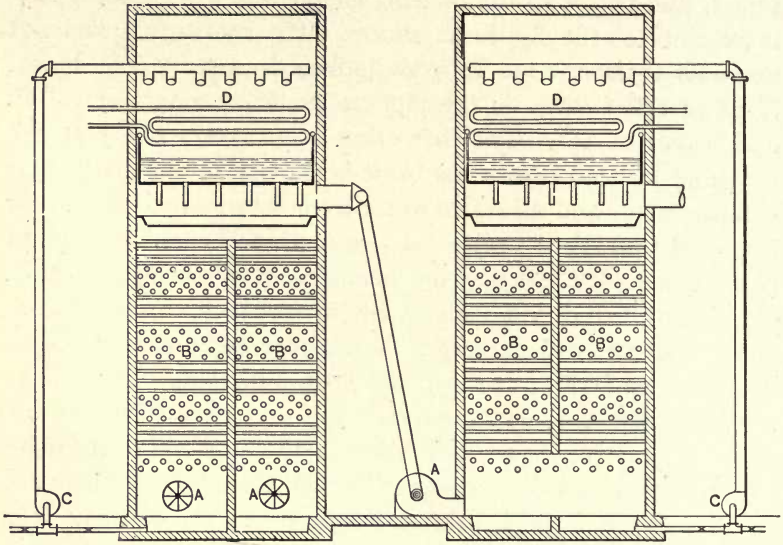


FIG. 171.—Gayley's Two-stage Air Cooler.

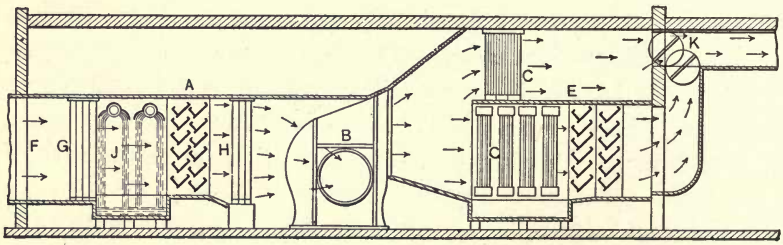


FIG. 172.—Air Conditioning Apparatus.

The **cooling of air for churches, hotels and auditoriums** or for rooms used in some manufacturing process is accomplished in the same way. In this case the air is freed from the precipitated moisture by first passing it through water to wash it and then over a set of baffle plates arranged as in *A*, Fig. 172, called eliminators for the purpose of removing the moisture.

The figure shows the arrangement of fan *B* and bunker *C*. The bunker *C* contains a number of pipes through which brine is passed to cool the air and precipitate the moisture. The eliminator *E* removes the moisture. The air enters at *F* and is passed through tempering coils *G* and *H* in cold weather. The washers *J* consist of a spray through which the air passes. This spray washes the air, taking out the dirt and gases. The upper coils *C* serve to warm part of the air if necessary. The mixing

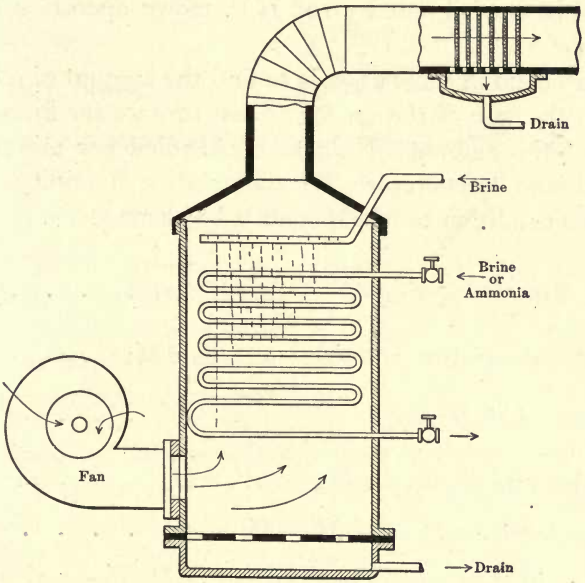


FIG. 173.—Bunker Room.

dampers at *K* are used to get a proper temperature of discharge.

The air may be cooled by passing it through a spray current of cold water or brine or the air may be passed over a set of revolving discs which dip into a cold-water or brine tank, and when they emerge they are cool and prepared to cool more air.

Fig. 173 illustrates another bunker room with vertical coils made up of horizontal pipes and return bends. These are connected to two mains. The coils are filled with brine.

At the Congress Hotel in Chicago a 300-ton machine is used in cooling the air and washing it for proper service.

At the Luther Memorial Church at Orange, Texas, a cooling plant is used to reduce the temperature from 90 and over to 70°.

The City Theatre of Rio Janeiro has recently been finished. This building seats 1700 persons with 200 on the stage. Over 50,000 cu.ft. of cold air per minute is introduced to bunker, cooling the air from 95° to 68°. This requires 7,000,000 B.t.u. to cool the air and 2,340,000 to condense the moisture. The operation is carried out by 105 H.P. motor operating an SO_2 compressor.

The problem in these cases is to find the amount of refrigeration. In the case of the air for a blast furnace the known data consist of the amount of air to be handled per minute, V_1, the maximum temperature T_1, the relative humidity of this ρ_1 and the condition to which it must be changed T_2.

$$\text{Weight of air entering} = \frac{(\text{Bar} - p_1\rho_1)V_1}{BT_1} = M_a \quad . \quad . \quad . \quad . \quad (8)$$

$$\text{Weight of moisture entering} = m_{1\rho 1}V_1 = M_1 \quad . \quad . \quad . \quad . \quad (9)$$

$$\text{Volume of air leaving} = \frac{M_a BT_2}{(\text{Bar} - p_2)} = V_2 \quad . \quad . \quad . \quad . \quad . \quad (10)$$

$$\text{Weight of moisture leaving} = m_2V_2 = M_2 \quad . \quad . \quad . \quad . \quad . \quad (11)$$

$$\text{Water condensed} = M_1 - M_2 = M_c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

$$\text{Energy in air above 32 entering} = 0.24\ M_a[T_1 - 491] = Q_1 \quad (13)$$

$$\text{Energy in moisture above 32 entering} = M_1[i_1] = Q_2 \quad . \quad (14)$$

$$\text{Energy in air above 32 leaving} = 0.24\ M_a[T_2 - 491] = Q_3 \quad (15)$$

$$\text{Energy in moisture above 32 leaving} = M_2[i_2] = Q_4 \quad . \quad (16)$$

$$\text{Energy in condensed moisture} = M_c q_2' = Q_5 \quad . \quad . \quad . \quad . \quad (17)$$

$$\text{Heat removed per minute} = Q = Q_1 + Q_2 - (Q_3 + Q_4 + Q_5) \quad (18)$$

m_1 = weight of 1 cu.ft. of saturated steam;
Bar = Barometric pressure;
p_1 = pressure of steam at temperature T_1;

T_1 = absolute temperature of entering air;
ρ_1 = relative humidity;
i_1 = heat content of moisture at entrance (superheated);
q' = heat of liquid.

For the cooling of buildings it is well to fix the temperature of the incoming air so that when heated to the temperature of the room it will have absorbed the heat entering into the room from the outside and from processes in the room. If the various heat losses through the walls be found from the K's of Chapter V,

Heat from walls $Q_e = \Sigma KF(t_a - t_r)$ . . . (19)

Heat from persons $Q_p = nQ'$ . . . . . (20)

Heat from machines and lights $Q_l = 2546 \times \text{H.P.} + Q_g n_1$. (21)

Now

$$Q_e + Q_p + Q_l = 0.02V(t_r - t_e) \quad . \quad . \quad . \quad . \quad (22)$$

0.02 = B.t.u. to heat 1 cu.ft. air 1° F;
V = volume of air per hour;
t_a = temperature outside air;
t_r = temperature room;
t_e = temperature of entering air.

V is fixed by the number of persons in the room. In some cases this is made 1800 cu.ft. per hour per person. This may be reduced to 1200 cu.ft. per sitting in an auditorium where the number is not fixed. The value of V is given by

$$V = 1200\, n. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

n = number of persons.

Having V, t_e may be found and the problem is the same from this point as the original problem for the blast furnace.

Having the heat removed from air the amount of surface required is given by

$$\text{Heat} = Q = FK \frac{\Delta t_1 - \Delta t_2}{\log_e \frac{\Delta t_1}{\Delta t_2}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

F = bunker surface in sq.ft;

$K = 2.2\sqrt{w_a}$ for wet surfaces;

$K = 1 + 1.3\sqrt{w_a}$ for dry surfaces;

Δt_1 = difference in temperature between air and brine at entrance or exit;

Δt_2 = difference in temperature between air and brine at exit and entrance.

The capacity of the refrigerating plant

$$\text{Tons of refrigeration} = \frac{Q}{199.2} \quad . \quad . \quad . \quad . \quad (25)$$

The brine cooler, condenser and compressor are fixed in the same manner as for any other problem.

This method may be used for the air needed and refrigeration for a chocolate factory.

Rinks. The use of expansion coils or brine coils for **skating rinks** has been employed in many places. The pipes are placed close together, using about 0.8 sq.ft. of brine pipe or 0.6 sq.ft. of direct-expansion pipe per square foot of rink. The heat to be removed is that from persons, walls, lights and fresh air.

Ice Cream. The making of ice cream has become a refrigeration problem of late years. The use of cold brine for the freezer in place of ice and salt was invented about 1902.

The cream when first received is stored in rooms or vats at a temperature of about 33° F. It is allowed to season for about twelve hours and after this it is put into a mixer in which the various ingredients are worked together. The mixture is now taken to the freezer, which may be of the batch or the continuous form. In the **Fort Atkinson Horizontal Freezer** of the Creamery Package Co., Fig. 174, a seamless German-silver cylinder with a scraper revolving against it has a brine coil of seamless copper pipe around it. The dasher of tinned bronze is caused to revolve in the opposite direction from the scraper. The whole tank is properly insulated. Above the cylinder is the feed tank to gauge the batch accurately. The

arrangement at the end of the feed tank permits one to put fruit in at this point without placing it in the main feed tank. In this freezer the mixture is discharged into the main cylinder and the dasher started. The cream is churned and cooled, and as the heat is removed there is a swell of about 69% of the volume, which occurs as the cream passes from 34°

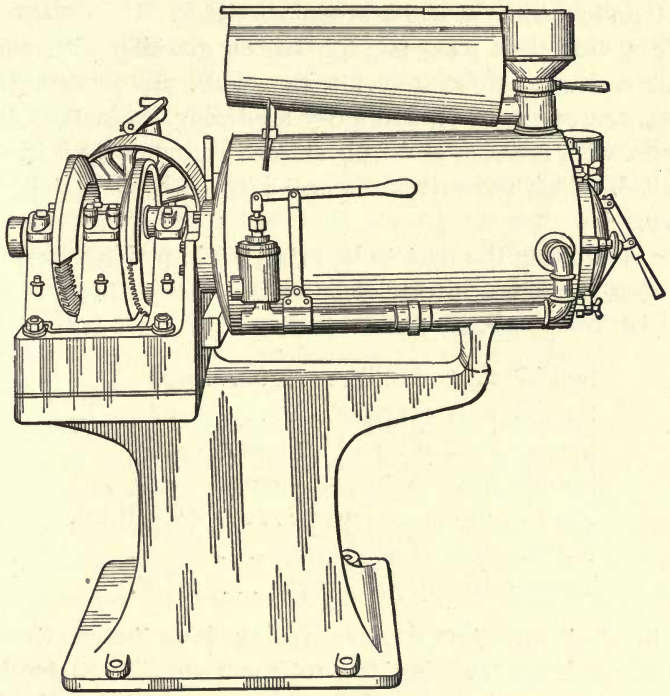


FIG. 174.—Fort Atkinson Freezer of Creamery Package Co.

to 28½° F. It depends on viscosity and the rate of freezing. The swell is due to air being introduced. After 23½° is reached the cream becomes brittle and the dasher will beat down the cream. The cream is not frozen hard in the freezer, but when the swell has occurred it is drawn off while it is still thin enough to flow slowly and is put into cans and fixed by storage. If the mixture is at 34° when introduced into the freezer it will be necessary to operate the dasher for from twelve to sixteen

minutes. The cream is now put in a hardening room at 0° F., where a fan keeps the air in circulation and thus removes the heat to harden the cream in six or eight hours. The old method of submerging the can in brine is not as good as the circulation method.

The freezers are made of various sizes, the 5-gallon size uses $\frac{3}{4}$ to 1 H.P. to drive, while a 10-gallon one takes $1\frac{1}{2}$ to 2 H.P., a 15-gallon, 3, and a 25-gallon, 5 H.P.

In a drug-store plant a 3-ton Larsen machine with 5-H.P. motor and a 40-quart freezer with 2-H.P. motor was placed in a space of 2 by 8 ft. and a dry hardening cabinet $3\frac{1}{2}$ by 18 by 3 ft. with a cream and fruit storage 10 by 12 by 8 ft. were installed. This shows what a small space is absolutely necessary.

In computing the heat to be extracted in making cream the following average figures may be used, although there is some variation from the various flavors of cream.

Specific heat of milk	0.90
Specific heat of cream	0.68
Specific heat of liquid ice cream	0.78
Specific heat of hard ice cream	0.42
Heat of fusion ice cream	80.0 B.t.u.
Temperature of hard cream	10° F.
Temperature of soft cream	16° F.

The first operation is the cooling from temperature of receipt, or if pasteurizing the cooling from the pasteurizing temperature to the storage temperature, after which the heat loss from the storage vat is cared for. The next cooling is from the mixing temperature to the temperature of 34° F., and then the heat to reduce the temperature to 28°, after which it is drawn out and stored. Part of the heat of fusion is taken out in the freezer and part in the hardening room. It may be assumed that one-half of the heat of fusion is removed in the freezer.

An important application of refrigeration is the **Poetsch process for sinking shafts**, first used in 1885. This is used

where a shaft has to be sunk through quicksand or where a soft wet stratum has to be penetrated. In these cases the side walls of the shaft would be forced out by the weight above, and to prevent this sheet piling or a caisson may be used, or when these are not possible the ground around the shaft is frozen. To do this a series of pipes is placed in holes left by a drill. This casing is put down and must penetrate the soft stratum. It is usually put down after a drill has bored a hole. By capping the end of the pipe and forcing water out of a small opening in the end of another pipe on one side of the large casing, this may be driven through the soft, sandy stratum by washing the sand before it. After the casing is put down, a separate pipe is put inside and then cold brine is pumped down and allowed to pass up through the annular space between the two pipes, removing heat from the damp earth, and freezing it into a solid wall, as shown by dotted lines in Fig. 175. In the Gobert system liquid ammonia is allowed to vaporize in a large casing, removing heat from the ground around.

To compute the amount of heat to be removed the temperature of earth must be found and its weight determined. The specific heat is 0.2. The amount of water to be frozen must be determined by drying out a sample of the earth of known volume and then the heat abstracted is found by using the general values of specific heats of water and ice and the heat of fusion of ice. The brine will freeze the earth for about 1 yd. from the pipe with the ring of pipes and half that distance on the outside with 0° brine, but the cooling will extend about 2 yds. beyond. This cooled layer represents the heat insulator and the heat carried across this area represents the heat which must be supplied to keep the ring frozen. The heat transmitted from zero brine amounts to about 85 B.t.u. per sq.ft. per hr., according to Lorenz.

The time taken to do this may be months, and for that reason a non-conducting house should be built around the top of the cooling pipes and the brine pipes must be carefully covered.

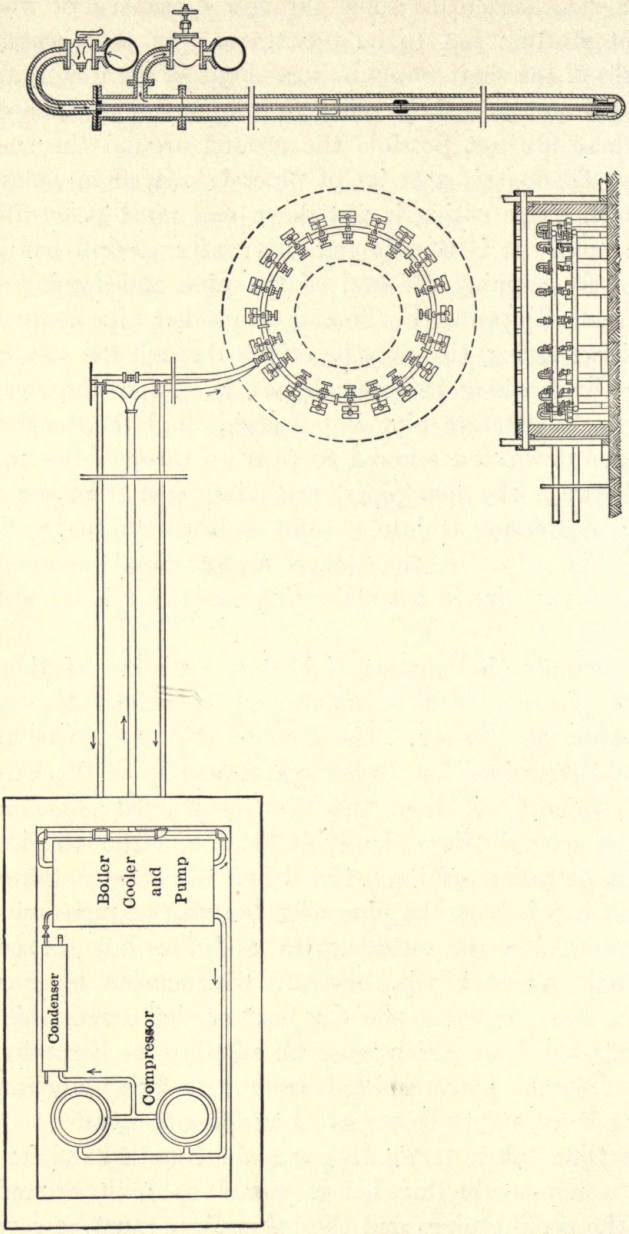


FIG. 175.—Poetsch System for Sinking Shafts

The sections of the heavy outer piping are joined on the inside, while the inner pipe is connected by ordinary couplings.

Another use for refrigerating machinery is the **cooling of drinking water.** This has been demanded in hotels by guests, and in factories it has been required by the manufacturer on account of the effect on the workman, by the workman for his bodily comfort, and by the legislature in laws for the bettering of working conditions. Of course this may be done by ice placed against cooling coils and ice put into the old-fashioned water cooler made of a barrel with a faucet, or the older pail and dipper, but the most hygienic method is to send filtered water through a brine or direct-expansion water cooler, then through a circuit of insulated pipe to the cooler again, taking off sanitary fountains at intervals.

Dr. Thomas Darlington states that about $3\frac{1}{2}$ pints of water should be drunk daily to care for water given off from the body. This water is necessary to aid digestion, to carry away waste and to properly regulate the actions of the body. The amount of water required varies with the amount of muscular exercise and with temperature. He states that the temperature should be about 50°, as ice water is apt to produce cramp and water that is not cool is so unpalatable that persons will not drink sufficient of it. At the National Tube Works water is cooled to 45° F. in summer and to 50° F. in winter. The water should not be carried in lead pipes, to avoid the danger of lead poisoning, and the endeavor should be made to filter the water to remove bacteria and sediment. Filtration makes the water attractive. The use of the drinking-cup common to all men should be discontinued, because of the easy transmission of disease thereby.

A large drinking-water cooling system has been installed by the National Tube Co. in Pittsburgh at their Continental Works.

The plant supplies fountains for about 1000 men, one fountain being used for each thirty men. These fountains must be located at convenient points, so that the men will drink, and so that the drinking will not consume too much time. The distri-

bution is made through 15,000 ft. of $1\frac{1}{2}$-in. galvanized steel pipe covered with $1\frac{1}{2}$ ins. of Nonpareil cork. The temperature rises about 7° in passing the circuit. The line loops down at each drinking-fountain, Fig. 176, as shown by the Nonpareil Cork Co. in their bulletin. In this way a continuous circuit of cold water is obtained so that there is always a discharge of cool water when the faucet is opened.

The water from the city filtration plant is first passed through two charcoal and gravel filters and then to a tank

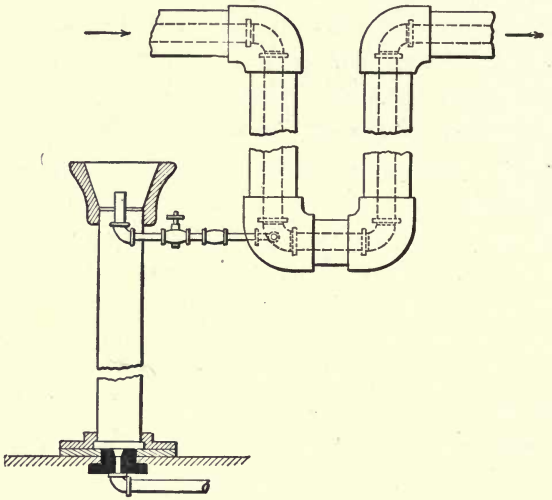


FIG. 176.—Drinking Fountain.

containing direct expansion coils, reducing the water to 45° in summer and 52° in winter. It is passed by means of a pump through three lines leading to all parts of the mill. The installation uses a 10-ton refrigerating machine for this plant. The amount of water, including waste, varies from about 1 gal. per man per day of ten hours in winter to $2\frac{1}{4}$ gallons in summer. The cost of this plant was $1.82 per employee per year against about $5 per man when ice and water tanks were used with the loss of men's time from sickness due to cold water. The system cost about $9000 to install.

In planning a system, $\frac{1}{4}$ gallon per hour per person should

be allowed to cover all wastes for summer use with hard muscular labor. In less active work this might be decreased to $\frac{1}{8}$ or $\frac{1}{10}$ gallon. In carrying this water a study must be made of the cost of pumping, which decreases with the size of pipe; the cost of heat loss through the insulation, which increases with the size of pipe; and the yearly cost of the covering and pipe for interest depreciation, taxes, and insurance, which

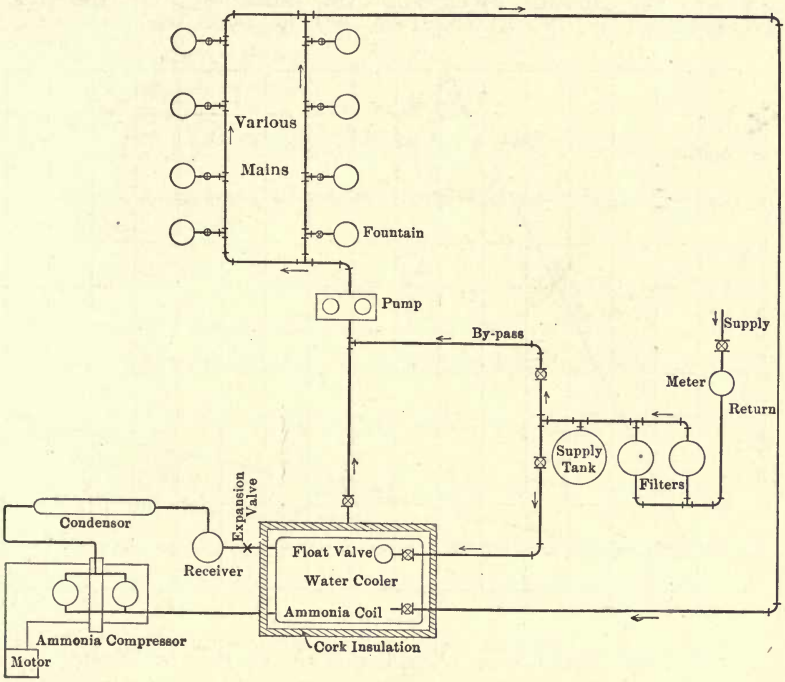


FIG. 177.—Diagram of Drinking Water Plant.

varies with the size of pipe. The first demands a large pipe, the second and third a small pipe. The yearly cost of several pipes should be figured, and that requiring the smallest cost used. The Nonpareil Co. recommend a velocity of about 3 ft. per second, which is considered in Chapter X. This figure may be used as a starting-point. A low velocity prevents the disturbance of any sediment in the pipe. The piping, of course, is arranged in a loop from the cooling tank back to

the tank. The fixtures are connected to the main flow pipe, as no dead ends are used. Long sweep elbows and bends will reduce the cost. The pipe may be covered by special ice-water pipe covering of Nonpareil cork $1\frac{1}{2}$ ins. thick and specially made for this service.

The heat required for such a system is made up of two parts: (*a*) the heat to cool the drinking water to 45° F. in summer and (*b*) the amount to care for the radiation from the pipe

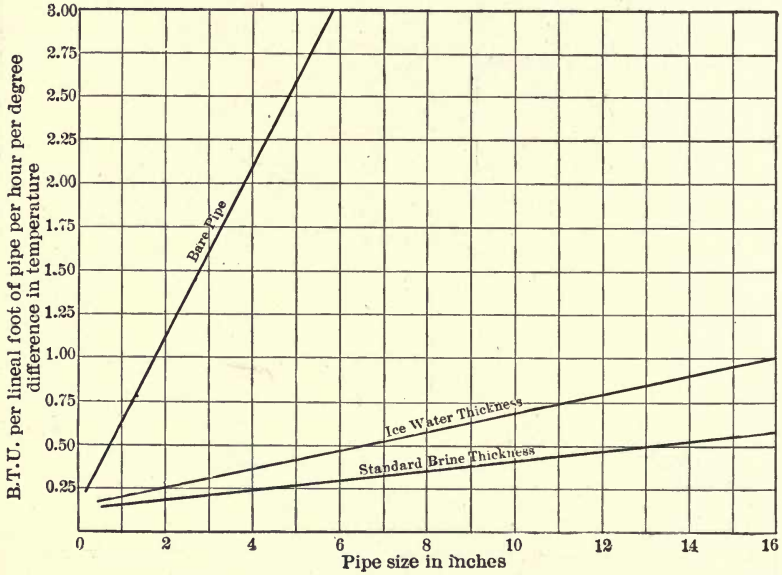


FIG. 178.—Heat Loss per Lineal Foot of Pipe per Hour per Degree.

covering. This latter should be such that the water is only warmed 7° in circulating through the pipe. This temperature is fixed by the length of circuit and the size of the pipe. The cork covering has been tested as shown in Chapter VI, and methods of that chapter may be used to compute the loss or 0.36 B.t.u. may be taken as the loss for 1 in. of thickness per square foot of cork surface at mean circumference per hour per degree difference in temperature. The loss from plain pipe is 0.8 B.t.u. per square foot per hour per degree dif-

ference. The heat loss is given in curve below for ice water thickness of cork, Fig. 178.

Water needed per hour

$$M_w = \frac{0.25 \times \text{No. of men at one time}}{7.48} \times 62.4. \quad (26)$$

Heat loss from pipe in any circuit $= H \times L$, . . (27)

H = heat loss per foot of pipe from curve for assumed diam. pipe;
L = length of pipe.

Heat loss in water flowing for 5° rise $= M_c \times 5$. . (28)

Hence the weight of water circulated to care for heat loss is given by

$$M_c = \frac{HL}{5}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

The area to allow for a 3 ft. per second velocity is given by

$$\tfrac{1}{2}M_w + M_c = F_p \times 3 \times 3600 \times 62.4 = 673{,}920F_p. \quad . \quad . \quad (30)$$

F_p = interior cross-sectional area of pipe in square feet.

The area thus found must check with that assumed for (27) and (29) and if it does not the assumed size must be changed.

Heat loss in pipes $= Q_p = \Sigma HL$ or $\Sigma M_c \times 5$. . . (31)

Heat to cool water entering $= Q_w = M_w(q_i - q_0)$.

q_i = heat of liquid at temperature of city supply (80°);
q_0 = heat of liquid at outlet temperature (45°) F.

Heat loss from tanks $= Q_t = FK(t_r - t_0)$. . . . (32)

F = area of surface of tank;
K = coefficient of transmission;
T_r = temperature of room;
T_0 = temperature of water in tank.

$$\text{Heat equal to work, } Q_f = \frac{(M_c + \frac{1}{2}M_w)h}{778}. \quad . \quad . \quad . \quad (33)$$

h = friction drop in system.

$$\text{Tons of refrigeration} = \frac{Q_v + Q_w + Q_t + Q_f}{60 \times 199.2}. \quad . \quad . \quad (34)$$

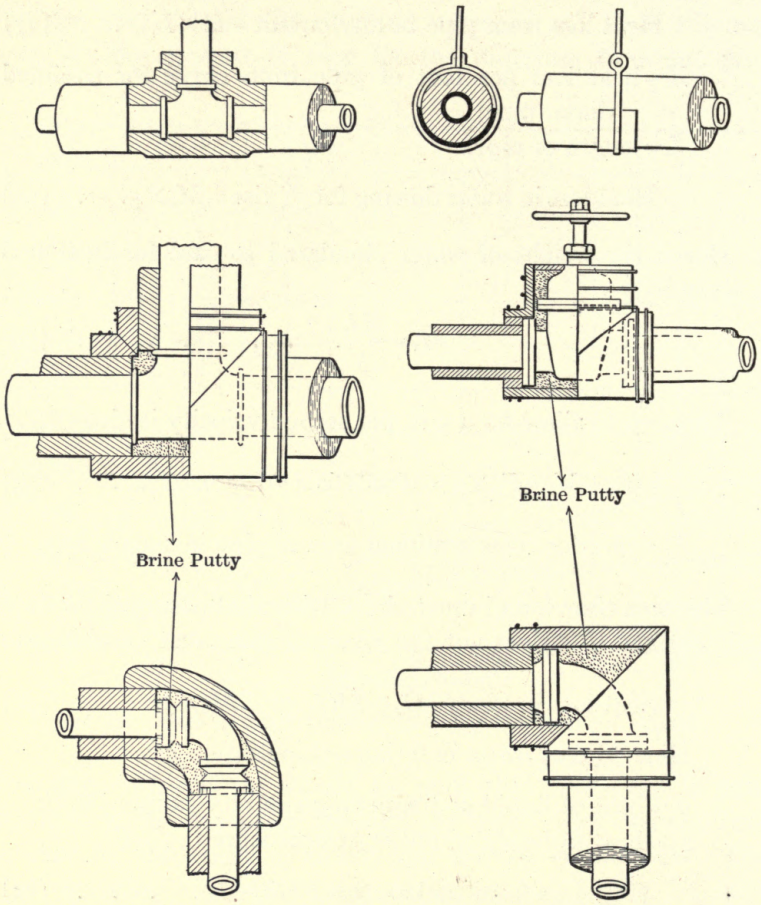


FIG. 179.—Methods of Covering Pipes and Fittings with Nonpareil Cork.

Fig. 179 illustrates the **section of cork covering** of various fittings and pipe, and Fig. 180 the insulation of an ice-water tank. This covering is made of boards of compressed cork

and to fit around pipes the cork is molded to form and where separated sections come together, a waterproof cement is used to make a tight joint. The sections are held together with four copper-covered steel wires. The pipe sectional coverings are put in so that the half sections break joints as shown. The outer surface is painted with an asphaltic paint and cavities are filled with brine putty or granulated cork and paraffin. Fig. 181 illustrates the arrangement of the ice-water plant of a large office building or hotel. The

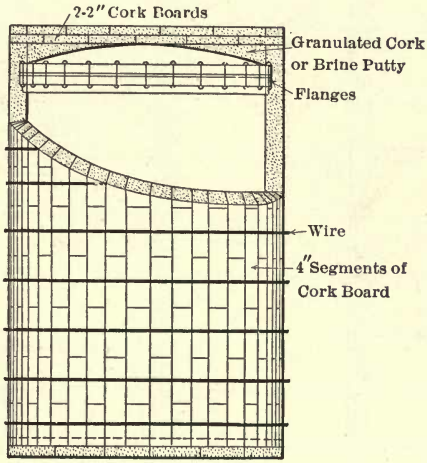


FIG. 180.—Armstrong Covering for Water Tank.

centrifugal pump, *A*, forces the water through the closed system. The filter *B* is used to supply fresh water to the cooler *C*, which is cooled by the direct-expansion coil or by brine around the water coil, which gives up its heat in the brine cooler. A closed system must be used in high buildings to balance the great static head, so that the pump will be required for friction only.

In **chemical works** the use of refrigeration to remove heat is similar to that for water cooling or chocolate making. There is nothing special in the methods of calculation. The quantities required are:

(*a*) Heat loss through walls.
(*b*) Heat from persons.
(*c*) Heat from motors.
(*d*) Heat of vaporization to condense vapors in process.
(*e*) Heat to cool liquids in process.
(*f*) Heat of fusion to solidify liquids in process.

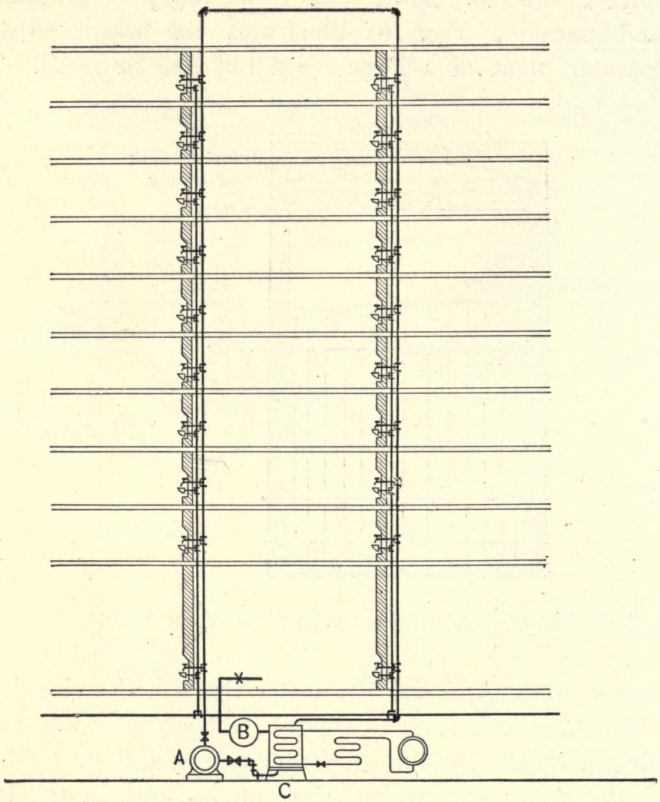


FIG. 181.—Drinking Water System in Hotel or Office Building.

The sum of these quantitites gives the heat to be removed and consequently the tonnage. The surface to abstract this heat is then found by fixing the temperatures on the two sides of a cooling surface and obtaining the coefficient of heat transfer. The problem is similar to any of the others.

The application of refrigeration to the **manufacturing of photographic supplies** and **to oil refining** has demanded large installation. Another use is to prevent chemical action by lowering the temperature of **ammunition holds of war vessels.**

The application of **refrigeration to the dairy** is shown in a cut from the Remington Machine Co. in Fig. 182. In this the apparatus used in a dairy is shown with the refrigerating machine near the office. The direct-expansion pipe used in the cold-storage room and in the cooler is not shown. The cold-

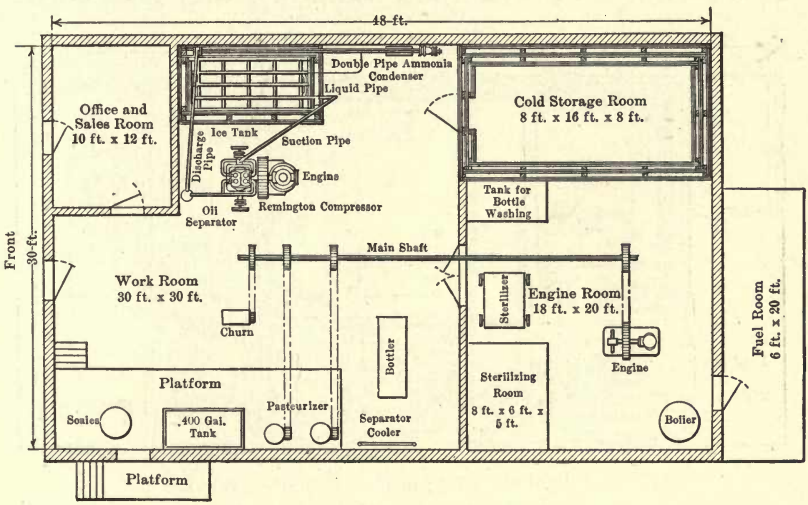


FIG. 182.—Complete Dairy Plant, 30′×48′. Remington Machine Co.

storage room is necessary to care for the milk and cream properly.

The apparatus of the Creamery Package Co. shown in Fig. 183 gives the requirements of the **modern creamery**. The ammonia compressor draws the ammonia from the brine cooler placed in the storage room or above it, the liquid being delivered to the cooler from the condenser by an expansion valve. The oil trap on the line from the compressor to the condenser is marked as well as the liquid receiver below the condenser. The brine pump circulates the brine from the brine tank to the pasteurizer and wizard back to the cold-storage

room. The milk is placed in the receiving vat and after reaching the proper temperature it is passed to the separator and from this the cream is passed to the pasteurizer and then to the wizard ripener, where it is allowed to age before being sent to the churn. It may be necessary to cool the cream in the pasteurizer or wizard, and for that reason these are connected by pipes to the brine system. For storage of butter, cream or milk a cold-storage room is used.

Another application is to the **manufacture of liquid air.** It is known that the throttling action of perfect gases occurs

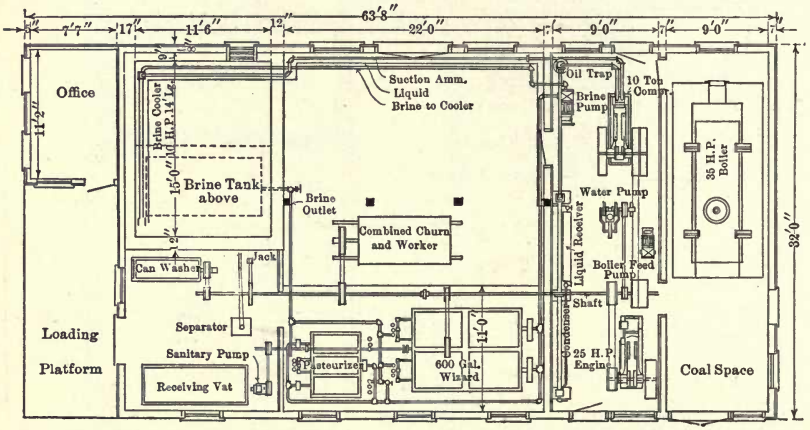


FIG. 183.—A Modern Creamery. Creamery Package Co.

at constant temperature because the heat content, which remains constant under such action, is a function of the temperature. However, there is no truly perfect gas and consequently when gases are throttled there is a slight drop in temperature known as the Thompson-Joule effect. Tripler, Hampson and Linde used this effect to obtain low temperatures. Linde machines have given the best results and are shown in Fig. 184. In this system a two-stage air compressor is used. One stage compresses atmospheric air to 240 lbs. per square inch pressure and the other stage to 3000 lbs. per square inch. The atmospheric air is compressed in the first stage and sent through a coil around the cylinder *A* placed in the jacket where it is cooled before going

to the second stage. The air there passes through an after cooler around the second stage *B*, after which it enters a separator *C* for oil and moisture. It then passes through a coil *D*, where it is cooled and then enters the inner pipe of a coil of three pipes *E*. In this coil the air is cooled by a current of low-pressure air which has been cooled to a low temperature, so that when the air reaches the end *F* of the coil it is quite cold. It is here allowed to expand from 3000 to 240 lbs. by the valve *G* and as a result its temperature should be lowered 203° C.

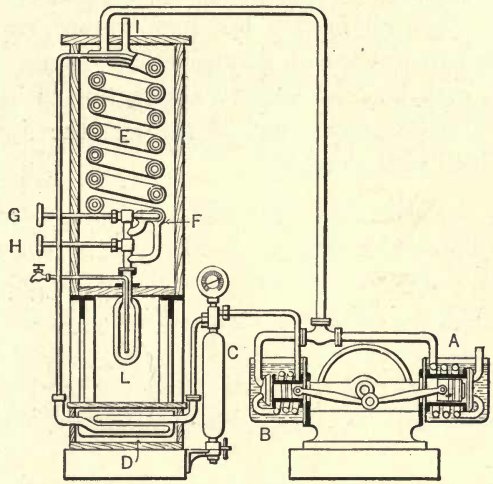


FIG. 184.—Linde Liquid Air Machine.

The incoming air could be cooled to 136.5° when the throttling is from 3000 lbs. to 240 lbs. per square inch absolute.

$$t_1 - t_2 = 0.276\left(\frac{3000}{14.7} - \frac{240}{14.7}\right)\left(\frac{273}{136.5}\right)^2 = 203^\circ * \quad . \quad . \quad (35)$$

Of course the air could not drop so much and the heat required to keep the heat content constant means that part of the air must be liquefied. Part of this air at 240 lbs. is throttled to 14.7 lbs. by *H* and is then sent out to the atmosphere through the outer annular space to *I*. The amount left between *G* and *H* is four-fifths of the total air, and this is sent

* Equation for Thompson-Joule effect.

back through the first annular ring. This air is at 240 lbs. per square inch and is taken to the intermediate receiver of the compressor. This air is cooled in the coil surrounding the cylinder *B* and the coil around *D* removes some of the heat from the high-pressure gas.

When the machine is started the air leaving at *G* and *H* may not liquefy, although there is a drop of 50° C. and this cools the next lot of gas, which of course drops to a lower temperature and soon liquid air appears.

In this apparatus the liquid air which forms is collected in the vessel *L*. The air is at a low temperature, corresponding to the boiling temperature at atmospheric pressure. These low temperatures may be used for any abstraction of heat to temperature at a little above that of the air, the liquid boiling away as the heat is abstracted.

CHAPTER IX

COSTS OF INSTALLATION AND OPERATION TESTS

THE cost of equipment, supplies, fuel and labor will vary from time to time and the figures given in this chapter have been collected, through the kindness of many manufacturers, as a guide to the student in determining cost of apparatus and manufacture. They should be used as guides only on account of the fluctuation in prices. They were compiled in 1916, but prices in use before the outbreak of the European war were employed.

Land. The cost of land will vary with the location in a city and with the city. In the outskirts of small towns it may be worth from 1 cent per square foot or $400 an acre to 5 cents a square foot or $2000 an acre. In a small city this will vary from $1000 an acre to $12,000 an acre, near the railroad. This latter price is about 30 cents per square foot. In the business districts of large cities $25 per square foot has been paid.

Buildings. The cost of buildings will vary with the type of structure. There are a number of variable units which enter into the problem and unit costs of various parts of a structure are given. For preliminary estimating the total cubic contents of the building, including cellar, may be found and then a unit cost selected from the table below is used to find the total cost. This is known as " cubing the building."

COST OF BUILDING PER CUBIC FOOT, UNINSULATED

Office Buildings

Frame	10 cts.	per cu.ft.,	$1.00	per sq.ft. floor
Brick and timber	13	"	1.25	"
Brick and steel	20	"	1.75	"
Reinforced concrete	20	"	1.75	"

Storehouses

Frame	6 cts. per cu.ft.,	$0.60 per sq.ft.	floor
Brick and timber	8 "	0.80 "	
Brick and steel	12 "	1.20 "	
Reinforçed concrete	12 "	1.20 "	

Power Houses

Frame	9 cts. per cu.ft.,	$0.90 per sq.ft.	floor
Brick and timber	11 "	1.10 "	
Brick and steel	15 "	1.50 "	
Reinforced concrete	15 "	1.50 "	

UNIT PRICES OF BUILDING ELEMENTS

Excavation and Hauling

Earth	$ .30 to .50 per cu.yd.
Rock	1.50 to 3.00 "

Masonry

Ordinary brick	33 cts. per cu.ft.,	$8.91 per cu.yd.	
Rubble stone	22 "	6.00 "	
1 : 3 : 5 concrete	22 "	6.00 "	
Reinforced concrete	37 "	10.00 "	
Concrete forms	$3.00 to $5.00 per cu.yd.		
Brick chimneys	$13.00 per cu.yd.		
Fireproofing	20 cts. per sq.ft.		
Steel Work	5 cts. per lb.		

Lumber

Heavy Georgia pine timber	$50.00 per M bd. measure
Georgia pine joist	40.00 "
Spruce joist	34.00 "
Yellow pine boards	25.00 "
Spruce boards	32.00 "
Ship lap, pine or spruce	26.00 "
Clapboards, pine or spruce	32.00 "
Cypress boards	60.00 "
Yellow pine flooring, vertical grain, $\frac{7}{8}''$	50.00 per M
Oak flooring, $\frac{3}{8}''$	70.00 "
Maple flooring	50.00 "
Shingles	2.50 to 5.00 per M
Lath (10 cts. per sq.yd. wall)	4.65 " "
Studding, $3''\times4''$ and $2''\times4''$ spruce	30.00 per M

Carpentering

Allow from one-half to full value of lumber for labor.

Plastering

Lime and hair	30 cts. per sq.yd.

Floors and Roadways

Asphalt facing, 2″	$1.20 per sq.yd.
Concrete sidewalks	$1.80 "
Concrete roadway, 6″	0.70 "
Macadam roadway, 6″	1.00 "
Brick roadway	1.75 "
Asphalt roadway	3.50 "
Concrete fireproof floors	18.00 per cu.yd.
	0.60 per cu.ft.

Partitions

Tiles 4″ thick (12″×12″)	5½ cts. per sq.ft.
8″ thick (12″×12″)	10 "

Labor equals cost of tile.

Roofing

Copper roofing	$25.00 per square (100 sq.ft.)
Slate roofing	$10.00 "
Tin roofing	7.50 "
Slag roofing	4.00 "
Book tiles, 2″	.07½ per sq.ft.
3″	.08½ "
Rain conductors, tin	.12 per ft.
Copper	.35 "

Mill Work

Windows with sash and trim	$8.00 to $12.00
Outer doors and frames	25.00 to 100.00
Inner doors and trim	8.00 to 15.00
Base boards	.08 to .16 per lin.ft.
Stairs	2.00 to 10.00 per step

Plumbing

Water-closets	$25.00 per unit
Wash basins	$12.00 per basin
Urinals	25.00 per stall
Soil pipe (iron)	.25 per ft.

Painting

White lead and oil	38 cts. per sq.yd. for 3 coats
Mineral paint	24 " " "
Asphaltum	35 " " "
Whitewash	15 " " " 2 "

Insulation

Building paper	$2.00 to $8.00 per roll of 500 sq.ft.
Asbestos, loose	$1.25 to $2.25 per 100 lbs., filling 3 cu.ft.
85% magnesia	$2.00 to $3.00 per 60 lbs., filling 3 cu.ft.
Hairfelt 1″ thick	.06 per sq. ft.

Cork boards: Walls. 2″ thick on brick or wood walls with cement finish, erected. 25 cts. per sq.ft.
2–2″ thicknesses, 40 cts. per sq.ft
1–3″ " 30 "
2–3″ " 60 "
Add 8 cts. for cork partition with two sides plastered.

Floors. 2″ cork board in asphalt, 3″ concrete top on asphalt covering with 1″ surface, 34 cts. per sq.ft. Same with 3″ cork, 40 cts. per sq.ft. 2–2″ layers, 50 cts. 2–3″ layers 60 cts.

Ceilings. 2″of cork on concrete or wood and ½″ cement plaster, 27 cts. per sq.ft.; 3″ cork, 32 cts.; 2–2″, 43 cts; and 2–3″, 64 cts.

Granulated cork: Unscreened granulated cork.. $70.00 per ton.
⅝ rescreened granulated cork... 60.00 "
⅜ granulated cork..... 35.00 "
Coarse regranulated cork..... 45.00 "
Fine regranulated cork..... 35.00 "

PIPE COVERING CORK (NET)

Size Pipe.	Cost per Foot.			Cost per Fitting.						
	Standard Brine Thickness.	Ice Water Thickness.	Cold Water Thickness.	Standard Screwed Fittings.			Standard Flanged Fittings.			
				Ells.	Tees.	Valves.	Ells.	Tees.	Valves.	Flanges.
¼	$0.34	$0.27	$0.24	$0.46	$0.50	$0.54	$2.20	$2.60	$3.05	$0.76
½	.43	.34	.30	.54	.63	.71	2.20	2.60	3.05	.76
1	.54	.43	.39	.71	.79	.87	3.05	3.50	3.90	.96
1¼	.63	.50	.45	.79	.88	.96	3.50	3.85	4.30	1.10
1½	.71	.57	.51	.88	.96	1.04	3.90	4.30	4.90	1.24
2	.80	.64	.57	.96	1.08	1.23	4.35	4.90	5.50	1.36
4	1.21	.97	.87	1.60	1.79	2.08	7.30	8.15	9.20	2.05
6	1.70	1.34	.....	2.00	2.30	3.02	11.10	12.05	13.50	3.05
10	3.40	2.70	.....	3.45	21.30	42.00	20.85	28.90	41.90	5.60
16	4.30	3.18	....	23.55	32.30	.....	39.75	55.90	71.00	6.80

PIPE COVERING, 85% MAGNESIA (NET)

Size Pipe.	Cost per Foot.		Cost per Fitting.		
	1½ Ins. Thick.	2 Ins. Thick.	Elbows.	Tees.	Valves.
1	$0.13	$0.21	$0.08	$0.09	$0.14
2	.16	.25	.09	.11	.15
3	.19	.29	.12	.14	.16
4	.22	.34	.15	.16	.38
6	.28	.43	.33	.40	.70
10	.42	.60	.90	1.15	1.55

Machinery Costs. These costs are made up of various items listed in the tables which follow. The prices represent average cost prices with discounts taken off. The items are for individual machines, but for complete equipment Mr. Thomas Shipley gives the following as a guide for the *cost of ice plants per ton of ice-making capacity* when they are at least of 50 tons capacity.

Compression can system	$550 per ton
Compression block system	650
Compression plate system (direct expansion)	800
Compression plate system (brine)	1000
Absorption can system	500

The yield of these plants will be $7\frac{1}{2}$ to 10 tons of ice per ton of coal in distilled-water can plants, 10 to 35 tons in raw-water can plants, and 10 to 15 tons in plate plants.

Refrigerating Plants. Cost of Mechanical Equipment:

Plants of 50 tons and over	$150 to $300	per ton of refrigeration
Plants of 8 to 20 tons	250	" "
Plants of 3 to 8 tons	300	" "
Plants of 1 to 3 tons	250	" "

Efficiency of Apparatus:

Boilers	60 to 80%
Producers	60 to 80
Steam engines (indicated thermal):	
Non-condensing:	
Simple	6%
Compound	10
Unaflow	11
Corliss	9
Condensing:	
Compound	20%
Mechanical efficiency of engines	85 to 95%
Steam turbines (overall thermal):	
Non-condensing	6%
Condensing, small	8
Condensing, medium	10
Condensing, large	21
Gas and oil engines:	
Indicated thermal efficiency	25 to 35%
Mechanical efficiency	85
Compressors:	
Mechanical efficiency	85 to 95%
Volumetric efficiency	88

Fuels:

Crude Oil:

Heating value per lb	19,000 to 20,000 B.t.u.
Weight per cu.ft	50 lbs.
Cost per barrel of 42 gallons	$1.50

Gasoline:

Heating value per lb	20,500 B.t.u.
Weight per cu.ft	50 lbs.
Cost per gallon	20 to 30 cts.

Bituminous coal:

Heating value per lb	13,800 B.t.u.
Weight per cu.ft., loose	50 lbs.
Cost per ton of 2240 lbs. at mine	$1.55
Cost of freight for 300 miles	1.90

Anthracite pea coal:

Heating value per lb	13,400 B.t.u.
Weight per cu.ft., loose	56 lbs.
Cost per ton of 2240 at mine	$2.65
Cost of freight, 200 miles	1.60

Anthracite buckwheat coal:

Heating value per lb	12,800 B.t.u.
Weight per cu.ft	56 lbs.
Cost per ton of 2240 lbs. at mines	$1.85
Cost of freight, 200 miles	1.50

BOILERS AND SUPERHEATERS. EFFICIENCY 65 TO 80%

Cost of Boilers

	Boiler Horse-power (10 Sq.ft. per H.P.).					
	50	100	200	300	400	500
Return tubular	$ 760	$1120	$2000	$2800		
Water tube	1500	2300	3600	4700	$5700	$7500
Superheaters, 10 to 15% of boiler surface for 100 to 120° F. superheat	600	600	1000	1300	1500	1600

PRODUCERS. EFFICIENCY 60 TO 80%

Cost of Producers

H.P.	80	100	150	200	250	300	400
Cost	$1600	$1800	$2200	$2500	$2800	$3200	$3800

Producers are based on 1.2 lbs. of coal per hr. per H.P. Grate areas of size to burn 9.4 to 10 lbs. of coal per sq.ft. per hour.

ENGINES AND TURBINES

Steam consumption of engines per I.H.P. hour:

Simple non-condensing	24 to 40 lbs.
Compound non-condensing	21 to 36
Compound condensing	14 to 20

COSTS

CORLISS ENGINES, SIMPLE (100 lbs. per sq.in. gauge)

Indicated H.P.	20	40	70	100	150	200	300
Size	8×18	10×30	12×30	14×36	16×36	18×42	22×42
Cost	$1000	$1200	$1500	$1900	$2150	$2700	$3500

CORLISS ENGINES, COMPOUND (125 lbs. per sq.in. gauge)

Indicated horse-power	100	600	1000
Size	10 and 18×36 Tandem	20 and 36×42 Cross	26 and 50×48 Cross
Cost	$3000	$10,000	$20,000

HIGH-SPEED ENGINES

Indicated horse-power	47 to 107	75 to 162	87 to 189	107 to 240	185 to 390
Size	10×10	12×12	13×12	14×14	18×18
Cost, belted	$755	$980	$1015	$1260	$2510
Cost, direct connected	1020	1270	1375	1617	3000

Piston speed from 550 to 650 ft. per min.
Steam pressure, 80 to 150 lbs. per sq.in. gauge.
Mechanical efficiencies, 85 to 95%.
Steam consumption, 29 to 35 lbs. per I.H.P. hour.

TURBO-GENERATORS

Capacity in K.W.	25 D.C.	100 D.C.	150 D.C.	200 D.C.	100 A.C.	200 A.C.
Cost	$1375	$3800	$5300	$6200	$4100	$5500

Steam consumption 40 lbs. per K.W. hr. in small sizes to 26 lbs. per K.W. hr. in large sizes. Both condensing.

GAS, GASOLINE, OIL OR PRODUCER ENGINES

Indicated horse-power..	5	10	25	50	100	200	300
Cost of gas or gasoline engine.....	$210	$380	$770	$1500			
Cost of fuel-oil engine..	.....	.....	1900	3100	4600	6800	9600
Cost of engine and suction producer.....	.....	.....	1800	2800	4700	7000	9800

Add 15% for freight and erection of engine and producer.

ELECTRIC GENERATORS (D.C.)

Efficiency 90 to 95%

Capacity in K.W.....	25	50	75	100	150	200
Cost, belted	$450	$600	$1000	$1000	$1500	$2200
Cost, direct connected	650	875	1150	1400	1850	2400

ELECTRIC MOTORS (D.C.)

Efficiency 85 to 95%

Horse-power.....	$7\frac{1}{2}$	15	25	50	75	100	150	200
Cost.....	$213	$290	$450	$605	$715	$1225	$1290	$2400

SWITCHBOARDS

SWITCHBOARDS FOR D.C. GENERATORS

Capacity in amperes.....	125	250	375	500	750	1000
Cost.....	$69	$78	$78	$87	$155	$175

Voltmeter, ammeter, rheostat, main switch and fuses.

SWITCHBOARDS FOR A.C. GENERATORS

Capacity in amperes.....	100	200
Cost.....	$125	$150

(Ammeter, voltmeter, exciter field switch, exciter and generator rheostat mounting, triple pole main switch and fuse.)

AMMONIA COMPRESSORS

Mechanical Efficiency 85 to 95%

Capacity in tons of ice. 2-ton refrigeration = 1 ton ice.	2	5	10	25	50	100	200
Cost of compressor (belt drive)	$550	$700	$1150	$1850	$3400	$8,700	$17,950
Cost of compressor and simple engine	730	1150	1800	2675	5000	10,670	21,160
Cost of compressor and compound engine	.....	.....	.....	4550	6500	13,750	27,500

AIR COMPRESSORS

Mechanical efficiency of compressor and motor 85%.
Efficiency of system from compressor motor to air motor 40%.

Free air in cu.ft. per min	55	110	250	350
Diam. steam cylinder, inches	6	8	10	12
Diam. air cylinder, inches	7	9	12	14
Stroke, inches	6	8	10	12
Price of engine compressor, governor and unloader	$520	$700	$1080	$1500
Price of belt-driven compressor with unloader	270	400	640	940
Max. pressure by gauge in lbs. per sq.in.	100	100	100	100

PUMPS

Direct acting for boiler feed, brine, or aqua ammonia.
Mechanical efficiency, 75%.
Steam per I.H.P. hour, 100 to 400 lbs.

Gallons brine per minute	100	250	500	1000
Size in inches for brine (Simplex)	6×6×7	8×8×13	12×12×20	
Cost	$170	$260	$570	$870
Weight in lbs	800	1660	4900	8650
Boiler horse-power	90	190	425	1000
Size in inches for boiler feed (Simplex)	5×2½×6	5×3½×7	6×4×12	9×6×13
Cost	$90	$110	$150	$250

CENTRIFUGAL PUMPS

Mechanical Efficiency 60%

Gallons per minute	100	250	500	1000
Speed	1800	1800	2000	1600
Horse-power	7.5	15	25	50
Weight in lbs	590	725	900	1500
Cost without motor	$180	$190	$270	$330
Pressure in lbs. per sq.in	100	100	100	100

AIR LIFT PUMP

Mechanical Efficiency 40%

FAN BLOWERS

Mechanical Efficiency 60%

Capacity at 1 oz. in cu.ft. per min.....	2000	4000	8000	16,000	26,000	40,000	54,000	72,000
Diam. wheel in inches.	15	21	30	41	53	64		
Cost.....	$120	$190	$270	$400	$600	$800	$1150	$1625

BELTING

Efficiency of Transmission 93 to 97%

Cost per inch of width, single thickness..... $8\frac{1}{2}$ cts. sq.ft.
Cost per inch of width, double thickness..... 17 cts. per ft.

Ammonia Condensers, Coils and Fittings. Allow 5 to 10° F. difference in temperature between water and vapor in saturated portion of condenser. Use 2-in. pipe for single-pipe condensers, 20 ft. long, and 24 pipes high in large stands. $1\frac{1}{2}$-in. pipe may be used. 3-in. and 2-in. pipe are used in double-pipe condensers. Use $K = 50$ for superheated portion of condenser, 100 to 200 for portion in which there is liquid on each side in double-pipe work; 60 is the value used in ordinary single pipe type of condenser. In the Block and Shipley forms of condenser $K = 200$. About 18 sq.ft. of surface is allowed per ton of capacity and it may be reduced to 8 sq.ft. where liquid ammonia is on the inner surface.

Cost of double-pipe condensers of $1\frac{1}{2}$- and 2-in. pipes 20 ft. long is given by:

Cost per stand = $5 + $15 × number of pipes high.

For 2 and 3-in. pipes, 20 feet long:

Cost per stand = $30 + $15 × number of pipes high.

Cost of Condensers with pan is given below:

Capacity in tons of ice.....	2	5	10	25	50	100	200
Cost for single-pipe condenser....	$160	$220	$400	$900	$1700	$3250	$6550
Cost for double-pipe condenser...	100	150	250	450	850	1700	3150

"DE LA VERGNE" STANDARD COUNTER-CURRENT ATMOSPHERIC AMMONIA CONDENSERS (FIG. 73)

Number of Stands.	Sq.ft. of Cooling Surface.	Capacity—Tons of Ice Melted per 24 Hours.	Length Over All.	Width Over All.	Sq.ft. of Floor Space.
1	222	12½	23′ 6″	4′ 0″	94
2	444	25	23′ 6″	6′ 0″	141
4	888	50	23′ 6″	10′ 0″	235
8	1776	100	23′ 6″	18′ 0″	423
16	3552	200	23′ 6″	34′ 0″	799
24	5328	300	23′ 6″	50′ 0″	1175

COST OF STEAM CONDENSER

Allow 1 sq.ft. for 5 lbs. of steam or design by Orrok's formula.

Shell type, sq.ft. surface	100	200	400	800
Cost	$250	$390	$725	$1280

Sheet iron type, 7′ 0″ high, 20′ 0″ long $190

Expansion Coils. Allow 275 sq.ft. of surface per ton of ice-making capacity in plate plants. Allow 300 lin.ft. of 1¼″ pipe in can tanks per ton of ice for ordinary coil and 200 to 250 lin.ft. when flooded.

Brine Coils. Allow 250 sq.ft. of surface per ton of ice in plate system.

Brine Coolers.

Tons of ice	2	5	10	25	50	100	200
Cost of double-pipe brine cooler	$ 64	$120	$250	$500	$1000	$1950	$3900
Cost of triple-pipe brine cooler	220	310	375	750	1500	3000	6000
Cost of shell-and-tube brine cooler	.....	150	200	360	650	1250	2500

Ammonia Separators. 24″×36″ welded $125

Ammonia Receiver. 12″×72″ welded 65

PIPING AND FITTINGS

[List price given.]

Size.	Pipe.		Ells.			Tees.		
	Std.	Ex. Hvy.	Std.	Ammonia.		Std.	Ammonia.	
				Screw.	Flange.		Screw.	Flange.
¼	$0.06	$0.07½	$0.05	$0.55	$1.05	$0.08	$0.80	$1.80
½	.08½	.11	.06	.65	1.30	.09	.95	2.10
¾	.11½	.15	.08	.75	1.60	.12	1.10	2.40
1	.17	.22	.10½	.90	2.10	.15	1.35	2.70
1¼	.23	.30	.16	1.20	2.45	.23	1.75	3.00
1½	.27½	.36½	.20	1.50	2.70	.29	2.30	4.50
2	.37	.50½	.28	1.90	3.25	.41	2.90	4.80
3	.76½	1.03	.75	2.90	7.50	1.10	4.35	12.60
4	1.09	1.50	1.20	5.10	20.00	1.75	7.80	31.00
6	1.92	2.86	2.75	11.40	40.00	4.00	17.10	48.00
Discount	75%	70%	70%	70%	70%	70%	70%	70%

Size.	Flanges.		Return Bends.			Valves.		
	Pairs.		Std.	Ammonia.		Std.	Ammonia.	
	Std.	Amm.		Screw.	Flange.		Screw.	Flange.
¼	....	$1.00	....	....	....	$1.01	$7.50	$7.40
½	0.40	1.20	....	....	....	1.60	8.50	8.40
¾	.46	1.60	$0.26	....	....	2.20	9.50	9.75
1	.52	1.80	.30	$0.95	....	2.80	10.50	11.15
1¼	.64	2.45	.40	1.30	2.80	4.00	13.00	17.95
1½	.78	2.95	.55	2.25	....	5.50	.....	19.45
2	1.00	3.60	.80	2.00	5.20	8.75	.....	23.10
3	1.50	6.00	2.20	....	....	12.50	.....	42.50
4	2.10	11.20	6.50	....	....	19.00	.....	84.00
6	3.95	20.00	....	....	....	37.50	.....	159.50
Discount	70%	70%	70%	70%	70%	60%	70%	70%

Wrought Iron Welded Steam, Gas and Water Pipe

(Table of Standard Dimensions)

Diameter.			Thickness, Inches.	Circumference.		Transverse Areas.			Length of Pipe per Sq.ft. of		Length of Pipe Containing 1 Cu.ft.	Weight per Foot Length, Pounds.	No. of Threads per Inch of Screw.	Contents in Gals. per Foot of Length,	Weight of Water per Foot of Length, Pounds.
Nominal Internal Inches.	Actual External Inches.	Actual Internal Inches.		External Inches.	Internal Inches.	External Sq. in.	Internal Sq. in.	Metal Sq. in.	External Surface Feet.	Internal Surface Feet.					
⅛	.405	.27	.068	1.272	.848	.129	.0573	.0717	9.44	14.15	2513.	.241	27	.0006	.005
¼	.54	.364	.088	1.696	1.144	.229	.1041	.1249	7.075	10.49	1383.3	.42	18	.0026	.021
⅜	.675	.494	.091	2.121	1.552	.358	.1917	.1663	5.657	7.73	751.2	.559	18	.0057	.047
½	.84	.623	.109	2.639	1.957	.554	.3048	.2492	4.547	6.13	472.4	.837	14	.0102	.085
¾	1.05	.824	.113	3.299	2.589	.866	.5333	.3327	3.637	4.635	270.	1.115	14	.0230	.190
1	1.315	1.048	.134	4.131	3.292	1.358	.8626	.4954	2.904	3.645	166.9	1.668	11½	.0408	.349
1¼	1.66	1.38	.14	5.215	4.335	2.164	1.496	.668	2.301	2.768	96.25	2.244	11½	.0638	.527
1½	1.9	1.611	.145	5.969	5.061	2.835	2.038	.797	2.01	2.371	70.66	2.678	11½	.0918	.760
2	2.375	2.067	.154	7.461	6.494	4.43	3.356	1.074	1.608	1.848	42.91	3.609	11½	.1632	1.356
2½	2.875	2.468	.204	9.032	7.753	6.492	4.784	1.708	1.328	1.547	30.1	5.739	8	.2550	2.116
3	3.5	3.067	.217	10.996	9.636	9.621	7.388	2.243	1.091	1.245	19.5	7.536	8	.3673	3.049
3½	4.	3.548	.226	12.556	11.146	12.566	9.887	2.679	.955	1.077	14.57	9.001	8	.4998	4.155
4	4.5	4.026	.237	14.137	12.648	15.904	12.73	3.174	.849	.949	11.31	10.665	8	.6528	5.405
4½	5.	4.508	.246	15.708	14.162	19.635	15.961	3.674	.764	.848	9.02	12.34	8	.8263	6.851
5	5.563	5.045	.259	17.477	15.849	24.306	19.99	4.316	.687	.757	7.2	14.502	8	1.020	8.500
6	6.625	6.065	.28	20.813	19.054	34.472	28.888	5.584	.577	.63	4.98	18.762	8	1.469	12.312
7	7.625	7.023	.301	23.955	22.063	45.664	38.738	6.926	.501	.544	3.72	23.271	8	1.999	16.662
8	8.625	7.982	.322	27.096	25.076	58.426	50.04	8.386	.443	.478	2.88	28.177	8	2.611	21.750
9	9.625	8.937	.344	30.238	28.076	72.76	62.73	10.03	.397	.427	2.29	33.701	8	3.300	27.500
10	10.75	10.019	.366	33.772	31.477	90.763	78.839	11.924	.355	.382	1.82	40.065	8	4.081	34.000
11	12.	11.25	.375	37.699	35.343	113.098	99.402	13.696	.318	.339	1.456	45.95	8	5.163	43.000
12	12.75	12.	.375	40.055	37.7	127.677	113.098	14.579	.299	.319	1.27	48.985	8	5.875	48.930

Wrought Iron Welded Extra Strong Pipe

(Table of Standard Dimensions)

Diameter.			Thickness Inches.	Nearest Wire Gauge Number.	Circumference.		Transverse Areas.			Length of Pipe per Square Foot of		Nominal Weight per Ft., lbs.
Nominal Internal Inches.	Actual External Inches.	Actual Internal Inches.			External Inches.	Internal Inches.	External Square. Inches.	Internal Square Inches.	Metal Square Inches.	External Surface Feet.	Internal Surface Feet.	
⅛	.405	.205	.1	12½	1.272	.644	.129	.033	.086	9.433	18.632	.29
¼	.54	.294	.123	11	1.696	.924	.229	.068	.161	7.075	12.986	.54
⅜	.675	.421	.127	10½	2.121	1.323	.358	.139	.219	5.657	9.07	.74
½	.84	.542	.149	9	2.639	1.703	.554	.231	.323	4.547	7.046	1.09
¾	1.05	.736	.157	8½	3.299	2.312	.866	.452	.414	3.637	5.109	1.39
1	1.315	.951	.182	7	4.131	2.988	1.358	.71	.648	2.904	4.016	2.17
1¼	1.66	1.272	.194	6½	5.215	3.996	2.164	1.271	.893	2.301	3.003	3.
1½	1.9	1.494	.203	6	5.969	4.694	2.835	1.753	1.082	2.01	2.556	3.63
2	2.375	1.933	.221	5	7.461	6.073	4.43	2.935	1.495	1.608	1.975	5.02
2½	2.875	2.315	.28	2	9.032	7.273	6.492	4.209	2.283	1.328	1.649	7.67
3	3.5	2.892	.304	1	10.996	9.085	9.621	6.569	3.052	1.091	1.328	10.25
3½	4.	3.358	.321	0	12.566	10.549	12.566	8.856	3.71	.955	1.137	12.47
4	4.5	3.818	.341	0	14.137	11.995	15.904	11.449	4.455	.849	1.	14.97
5	5.563	4.813	.375	00	17.477	15.120	24.306	18.193	6.12	.687	[illegible]3	20.54
6	6.625	5.75	.437	000	20.813	18.064	34.472	25.967	8.505	.577	.6[illegible]4	28.58

Double Extra Strong Pipe

½	.84	.244	.298	1	2.639	.766	.554	.047	.507	4.547	15.667	1.7
¾	1.05	.422	.314	1	3.299	1.326	.866	.139	.727	3.637	9.049	2.44
1	1.315	.587	.364	00	4.131	1.844	1.358	.271	1.087	2.904	6.508	3.65
1¼	1.66	.885	.388	00	5.215	2.78	2.164	.615	1.549	2.304	4.317	5.2
1½	1.9	1.088	.406	000	5.969	3.418	2.835	.93	1.905	2.01	3.511	6.4
2	2.375	1.491	.442	0000	7.461	4.684	4.43	1.744	2.686	1.608	2.561	9.02
2½	2.875	1.755	.560	1/16 −	9.032	5.513	6.492	2.419	4.073	1.328	2.176	13.68
3	3.5	2.284	.608	3/16 −	10.996	7.175	9.621	4.097	5.524	1.091	1.672	18.56
3½	4.	2.716	.642	5/16 +	12.566	8.533	12.566	5.794	6.772	.955	1.406	22.75
4	4.5	3.136	.682	1/16 −	14.137	9.852	15.904	7.724	8.18	.849	1.217	27.48
5	5.563	4.063	.75	3/16	17.477	12.764	24.306	12.965	11.34	.687	.940	38.12
6	6.625	4.875	.875	3/16	20.813	15.315	34.472	18.666	15.896	.577	.784	53.11

Piping for Ice Storage Rooms:

Capacity in tons of ice	2	5	10	25	50	100	200
Cost	$75	$100	$150	$200	$350	$450	$750

CANS AND DISTILLING APPARATUS

Allow twelve 300-lb. cans per ton for flooded system, otherwise fourteen 300-lb. cans.

Capacity in tons of ice	2	5	10	25	50	100	200
Cost of cans and tank with coils, raw water	$950	$1400	$1950	$3850	$6850	$13,250	$26,100
Cost of cans and tank with coils, distilled water	550	900	1600	3200	5750	11,400	22,900
Distilling apparatus	400	500	600	1000	1650	2,500	4,400

Reboilers, 60″×60″ $100
Skimmers, 20″×120″ 40
Water coolers of 1½ and 2 ft. pipe, 20 ft. long. Same as double-pipe condensers.
Sponge filters, 9″×42″ $35

MISCELLANEOUS APPARATUS AND SUPPLIES

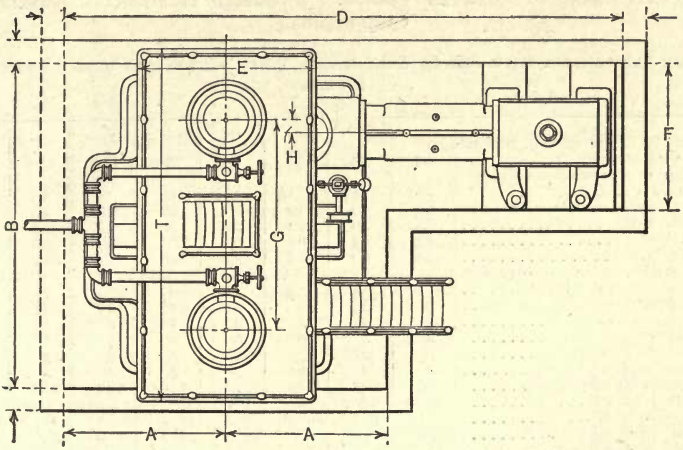
Agitators. 12″ belt driven ..... $ 36
8″ motor and agitator 200
Can Fillers: For 300-lb. cans $15
Can-filling hose ..... 40 cts. per ft.
Can hoists, 300-lb. cans:
Electric $500
Air 200
Hand 75
Ice dump, 300-lb. single 50
300-lb. double 90
Ammonia ..... 25 to 30 cts. per lb.
Carbon dioxide ..... 5 cts. per lb.
Sulphur dioxide ..... $1 per lb.
Calcium chloride ..... $14 per ton
Sodium chloride ..... $14 per ton
Water from city ..... 5 to 20 cts. per 1000 gal.

Data from Vertical York Compressors

With 185 Lbs. (95.5° F.) Head Pressure and 15.67 Lbs. (0° F.) Suction Pressure.

Tons, Capacity.	Size of Cylinders.		Cubic Inches Displacement Ammonia Cylinders and Horse Power.		Weight Fly-wheel Diam.	R.P.M.	A	B	D	E	F	G	H
	Ammonia.	Steam.											
1¼	1–4¾ × 5	5 × 5	80	.....	30″ 400	131	.....	.....	5′ 3″				
3	1–6 × 6	6 × 6	169	.....	36″ 500	148	.....	.....	5′ 6″				
6	2–6 × 6	8 × 6	339	.....	36″ 600	148	.....	.....	5′ 9″				
10	2–7½ × 10	11½ × 10	872	.....	48″ 2200	96	.....	.....	7′				
20	2–9 × 12	13½ × 12	1,526	35	54″ 1900	110	3′ 6″	8′ 0″	10′ 6″	.....	4′ 6″	4′ 10″	3″
35	2–11½ × 15	16 × 15	3,114	60	60″ 2400	94	3′ 9″	8′ 9″	11′ 6″	.....	4′ 9″	5′ 4″	3½″
65	2–14 × 21	20 × 21	6,464	111	7′ 0″	84	5′ 6″	10′ 11″	18′ 0″	.....	4′ 9″	7′ 0″	5″
90	2–16 × 24	24 × 24	9,648	154	8′ 0″	78	5′ 8″	12′ 6″	20′ 2″	7′ 0″	5′ 9″	8′ 2½″	5¼″
125	2–18 × 28	26 × 28	14,246	214	9′ 0″	74	6′ 0″	12′ 6″	22′ 0″	7′ 10″	5′ 10½″	8′ 5″	5½″
175	2–21 × 32	28½ × 32	22,163	299	10′ 0″	66	7′ 2	14′ 6″	24′ 8″	7′ 10″	6′ 6″	9′ 4″	6½″
250	2–24 × 36	34 × 36	32,566	427	12′ 0″	64	8′ 3″	16′ 6″	28′ 2″	8′ 3½″	7′ 9″	10′ 9″	8″
350	2–27 × 42	36 × 42			16′ 0″								
503	2–30 × 48	30 & 58 × 48	67,853	855		62	11′ 0″	21′ 0″	37′ 0″	11′ 8″	10′ 0″	13′ 7¾″	9″

Tons, Capac-	I	J	K	L	M	N	O	P	T	S	U	V	W	X	Y
1¼	.....	.....	.....	5′ 3″	1	1¼	.....	.....	2′ 9″	.....	.....	.....	1″	1″	6′ 0″
3	.....	.....	.....	5′ 6″	1¼	1½	.....	.....	3′ 9″	.....	.....	.....	1¼″	1¼″	7′ 0″
6	.....	.....	.....	5′ 9″	1½″	2″	.....	.....	4′ 9″	.....	.....	.....	1¼″	1¼″	7′ 0″
10	.....	.....	.....	7′ 9″	2″	3″	.....	.....	5′ 6″	.....	.....	.....	1½″	1½″	9′ 6″
20	20¾″	25″	17″	9′ 3½″	2½″	3″	5 0¼″	4′ 0″	6′ 6″	6′ 2¼″	.....	22⅛″	1½″	2″	12′ 0″
35	2′ 0″	2′ 2¼″	10⅛″	11′ 0″	3	3½	5′ 10⅝″	4′ 0″	7′ 3″	7′ 2⅜″	.....	2′ 6″	2″	2½″	14′ 6″
65	2′ 0⅝″	2′ 8¼″	23⅓″	16′ 2⅝″	4½″	6″	9′ 6½″	4′ 6″	12′ 2″	9′ 6⅛″	13′ 10″	3′ 7⅝″	3½″	3½″	
90	2′ 5″	3′ 1″	2′ 3½″	17′ 11″	4½″	6″	10′ 9½″	4′ 6″	14′ 2″	11′ 3″	15′ 7⅝″	3′ 4½″	3½″	4″	20′ 0″
125	2′ 8″	3′ 2″	2′ 5″	20′ 4½″	5″	7″	12′ 6½″	5′ 0″	14′ 6½″	12′ 5″	17′ 3⅜″	3′ 10½″	3½″	5″	22′ 0″
175	2′ 11″	3′ 7¼″	2′ 8⅜″	23′ 0″	7″	8″	14′ 1⅝″	5′ 0″	15′ 7″	14′ 0″	19′ 11½″	4′ 10¾″	5″	5′	24′ 6″
250	3′ 3½″	4′ 3″	2′ 10¾″	26′ 1½″	8″	10″	15′ 10⅞′	5′ 0″	17′ 2½″	15′ 7″	22′ 2½″	5′ 3½″	5″	6″	27′ 0″
503	4′ 5″	5′ 3″	3′ 6½″	34′ 3″	10″	12″	20′ 11½″	10′ 0″	19′ 4″	21′ 2 3/16″	29′ 9″	6′ 6″	7″	10″	39′ 0″



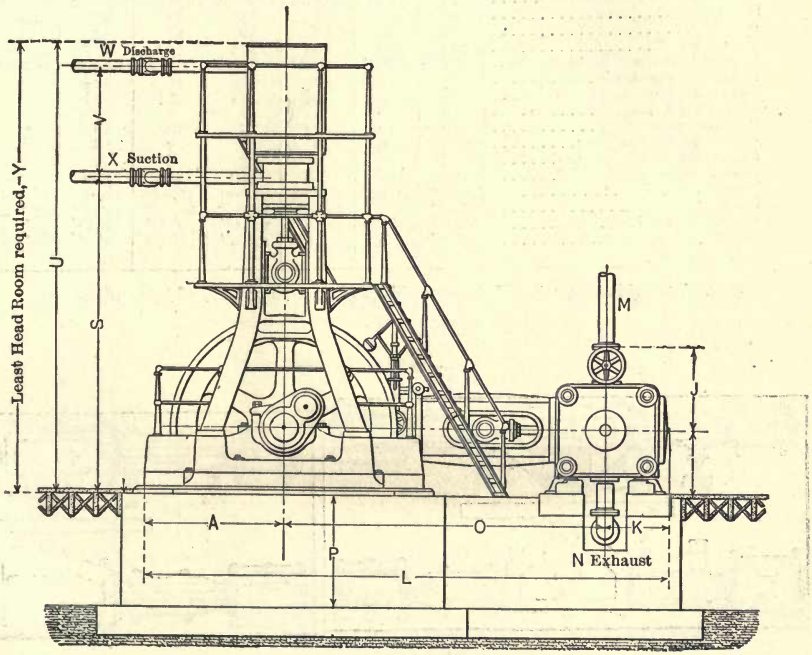


FIG. A.—York Compressor.

Particulars of "De la Vergne" Standard Horizontal Ammonia Compressors

One Compressor with Simple Engine and Two Compressors with Compound Engine

Capacity, tons of ice melted per 24 hours *	25	35	50	75	100	125	150	200
Diameter compressor cylinder, ins.	10½	11½	13½	14½	16	18	20	22
Diam. steam cylinder, ins.	15	17	19	20	22	26	26	32
Stroke, ins.	18	20	22	26	30	33	33	36
R.P.M.	63	65	62	68	65	58	56	57
Rated H.P.	45	60	85	130	170	215	255	340
Dimensions main bearings, ins.	6½ × 11	7 × 12½	8 × 15	9 × 15	10 × 18	11 × 20	12 × 22	13 × 24
Diameter crank pin, ins.	4¼	4½	5¼	5½	6	7	7½	8½
Diameter cross-head pin, ins.	3	3½	4	4½	4½	4¾	5½	5½
Steam pipe, ins.	3	3	4	4	5	6	6	7
Exhaust pipe, ins.	4	4	6	6	6	7	8	9
Ammonia suction, ins.	2	2½	3	3	4	4	5	5
Ammonia discharge, ins.	2	2	2½	3	3	4	4	5
Diameter flywheel, ins.	96	105	120	128	136	144	160	160
Weight, do, lb.	5000	6000	7000	9000	10,500	14,000	17,000	19,000
Length over all	13′ 10″	15′ 1″	16′ 3″	18′ 5″	21′ 2″	22′ 7″	24′ 7″	27′ 3″
Width over all	8′ 6″	8′ 9″	9′ 6″	11′ 0″	11′ 2″	12′ 6″	12′ 6″	15′ 11″
Height above floor	6′ 0″	6′ 5″	7′ 1″	7′ 7″	8′ 2″	9′ 3″	9′ 5″	10′ 0″

Capacity, tons of ice melted per 24 hours *	250	300	250	300	400	500	600
Diameter compressor cylinder ins.	24	26	2–18	2–20	2–22	2–24	2–26
Diam. steam cylinder, ins.	34	36	24 & 48	27 & 54	29 & 58	32 & 64	34 & 68
Stroke, ins.	40	48	33	33	36	40	48
R.P.M.	54	45	58	56	57	54	45
Rated H. P.	425	510	440	525	700	875	1050
Dimensions main bearings, ins.	14 × 28	16 × 28	14 × 28	15 × 28	17 × 30	18 × 34	20 × 36
Diameter crank pin, ins.	9	9½	12	12½	13	13½	15
Diameter cross-head pin, ins.	5½	9	4¾	5½	5½	5½	9
Steam pipe, ins.	7	8	5	6	7	7	8
Exhaust pipe, ins.	10	10	14	15	16	18	20
Ammonia suction, ins.	6	6	6	6	7	8	9
Ammonia discharge, ins.	5	6	5	5	6	7	8
Diameter fly-wheel, ins.	174	192	160	160	174	192	216
Weight, do. lb.	23,000	40,000	19,000	23,000	26,000	40,000	50,000
Length over all	20′ 0″	33′ 0″	34′ 5″	38′ 0″	40′ 2″	43′ 4″	47′ 4″
Width over all	16′ 6″	17′ 6″	12′ 6″	12′ 6″	12′ 8″	13′ 0″	13′ 11″
Height above floor	10′ 7″	11′ 0″	10′ 8″	11′ 0″	11′ 7″	13′ 0″	12′ 11″

* The ice-making capacity of these machines is from 50 to 60% of this rating.

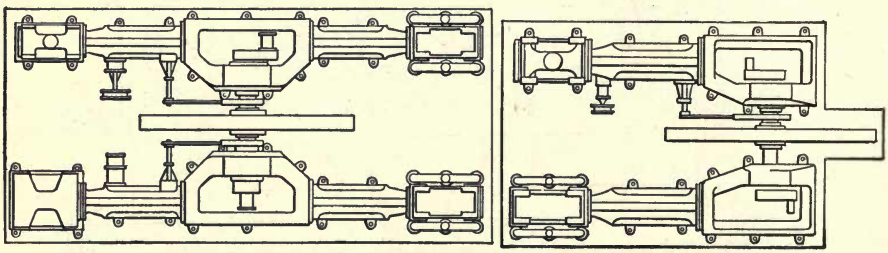


Fig. B.—De La Vergne Compressors.

Brunswick Refrigerating and Ice Making Machines

COMPRESSOR DATA

Number.	Tons Refrigerating Capacity, 24 Hours.	Belt-driven Compressors.														Compressors with Steam Engine Direct Connected							
						Fly-wheel.				Size Pipe Connections.		Space Required for Machine.			Net Weight.								
		Number of Cylinders.	Diameter of Cylinder, Inches.	Stroke, Inches.	Revolutions per Minute.	Diameter, Inches.	Face, Inches.	Weight, Pounds.	Diameter of Shaft, Inches.	Suction, Inches.	Discharge, Inches.	Length, Inches and Feet.	Width, Inches and Feet.	Height, Inches and Feet.	Compressor Only, Pounds.	Compressor and High Side Complete, Pounds.	Steam Engine.			Space Required for Compressor and Engine.			Net Weight.	
																	Diameter of Cylinder, Inches.	Stroke, Inches.	Exhaust, Inches.	Length, Inches and Feet.	Width, Inches and Feet.	Height, Inches and Feet.	Compressor and Engine Only, Pounds.	Direct Connected Compressor and High Side Complete, Pounds,
C—¼	¼	1	2	2	275–300	16	2½	55	1 3/16	¼	¼	–22	–16	–36	225	500	3	3	1	–36	–16	–36	590	815
C—½	½	1	2½	2¾	250–275	21	3⅜	135	1½	½	½	–35	–25	4–2	485	975	4	4	1	4–6	–25	4–6	925	1,395
C—1	1	1	3½	3½	190–210	25	5	310	1 15/16	¾	½	–38	–25	5–0	1085	1850	5	5	1	4–8	–25	5–7	1670	2,370
C—2	2	1	4½	4½	170–180	20	5½	475	2 7/16	1	¾	–42	–30	5–10	1590	2710	5½	7	1¼	5–0	–30	6–0	2350	3,340
C—4	4	1	6	6½	150–160	34	6½	655	2 15/16	1¼	1	4–2	–34	6–4	2100	3770	6½	7	1½	6–4	–34	6–7	3400	4,920
C—6	6	1	7½	8	130–140	40	8½	1145	3 7/16	1½	1¼	5–0	–40	7–3	4130	6465	8	8	2	7–4	–40	7–6	5240	7,425
C—8	8–10	2	6	6½	150–160	42	10	1120	3 15/16	1½	1¼	5–3	–45	7–1	4540	7270	9	9	2½	7–10	–42	8–0	5880	8,530
C—12	12–15	2	7½	8	130–140	52	12½	1700	3⅞	1½	1¼	6–0	4–4	7–8	5600	9715	10	12	3	8–5	4–4	8–6	7200	11,145

Note. Complete ¼-ton compression side with automatic expansion valve mounted on pedestal, ready for connection to coils, overall dimensions: 32 ins. long, 22 ins. wide, 38 ins. high.

DATA FROM HIGH-SPEED ENGINE—ERIE CITY IRON WORKS

Size of Engine.	*A*	*B*	*C*	Weight of Fly-wheel.	*D*	*E*	*F*	*G*	*H*	*I*
	Ins.	Ins.	Ins.		Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
10 × 10	4	48	9½	1345	15½	18½	36	28	75	103
11 × 10	4	48	9½	1345	15½	18½	42	30	75	105
12 × 12	4¾	54	12½	2025	18¼	22¼	48	30	90	120
13 × 12										
14 × 14	5½	60	14½	2920	21¼	26⅛	54	36	105	131
15 × 14										
16 × 16	7	72	16½	4500	24⅛	29⅛	60	40	120	160
17 × 16										
18 × 18	8	72	18½	5200	27¾	39½	66	48	131	179
19 × 18										

Size of Engine.	*J*	*K*	*L*	*M*	*N*	Speed.	Initial Steam Pressure.	I.H.P.
	Ins.	Ins.	Ins.	Ins.	Ins.			
10 × 10	85	25½	54½	99	51	300 to 350	80 to 150 lbs.	47 to 104
11 × 10	91	25½	60½	99	51	300 to 350		57 to 126
12 × 12	108½	30¾	72¼	117	61½	275 to 325		75 to 162
13 × 12						275 to 325		87 to 189
14 × 14	130¼	35⅝	80⅛	135	71½	250 to 300		107 to 240
15 × 14						250 to 300		123 to 275
16 × 16	134¼	40⅝	89⅛	150	81¼	225 to 275		146 to 333
17 × 16						225 to 275		165 to 370
18 × 18	148	47¼	105½	167	92½	200 to 225		185 to 390
19 × 18						200 to 225		205 to 435

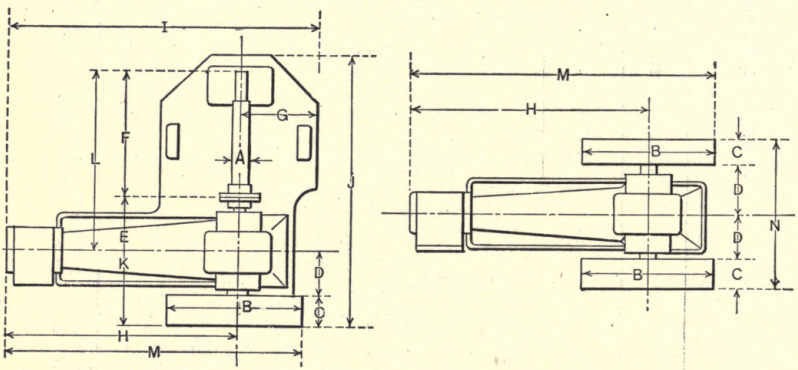


FIG. C.—Erie City Iron Co. Engines.

Data for Turbo Generators

Power.	R.P.M.	Width.	Length.	Limit Length.	Height.	Steam Pipe.	Exhaust Pipe.	
K.W.		Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ins.	Ins.	
35	D.C. 3600	2 10	6 6	7 11	3 $0\frac{7}{8}$	2	$4\frac{1}{2}$	Non-condensing
100	A.C. 3600	4 10	12 $4\frac{3}{4}$	12 $11\frac{3}{4}$	4 9	3	8	"
200	D.C. 3600	4 $4\frac{7}{16}$	12 $4\frac{1}{2}$	12 $4\frac{1}{4}$	4 $11\frac{1}{2}$	$3\frac{1}{2}$	8	"
200	A.C. 3600	5 0	11 $9\frac{3}{8}$	13 $10\frac{3}{4}$	4 9	$3\frac{1}{2}$	8	"

Data for Return Tubular Boilers

Boiler H.P.	Shell.		Tubes.		A	B+K	C	C'	D	E
	Diam.	Length.	Diam.	No.						
	Ins.	Ft.	Ins.		Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.
46.5	42	15	3	34	42	16 2	2 2	1 11	8 5	7 9
65.4	48	15	3	50	48	16 2	2 2	1 11	8 11	8 1
80.1	54	15	3	62	54	16 2	2 2	2 2	9 6	8 9
98.7	60	16	3	72	60	17 2	2 3	2 1	10 0	9 1
131.9	66	18	$3\frac{1}{2}$	74	66	19 6	2 3	2 3	10 8	9 8
167.5	72	18	$3\frac{1}{2}$	96	72	19 6	2 3	2 3	11 2	10 0
197.1	78	18	$3\frac{1}{2}$	114	78	19 8	2 6	2 6	12 7	10 10
242.0	84	18	$3\frac{1}{2}$	142	84	19 8	2 6	2 6	12 10	11 3
283.0	90	18	$3\frac{1}{2}$	168	90	19 10	2 6	2 6	13 4	11 8
319.7	96	18	$3\frac{1}{2}$	190	96	19 10	2 6	2 6	13 10	12 1
354.6	96	20	$3\frac{1}{2}$	190	96	21 10	2 6	2 6	13 10	12 1
391.6	96	20	3	248	96	21 10	2 6	2 6	13 10	12 1

Boiler H.P.	F	G for 1 Boiler.	H	I	J	K	L	M	N	c to c Shells in Battery.	Red Brick.	Fire Brick.
	Ft. Ins.	Ft. Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ft. Ins.		
46.5	19 2	8 0	$33\frac{1}{4}$	$40\frac{1}{2}$	2	43	37	3	48	5–8	15,000	1300
65.4	19 2	8 6	$33\frac{1}{4}$	$48\frac{1}{2}$	6	49	43	4	54	6–2	16,000	1525
80.1	19 2	9 0	$33\frac{1}{4}$	$48\frac{1}{4}$	6	55	49	4	60	6–8	16,700	1650
98.7	20 2	9 6	$33\frac{1}{4}$	$48\frac{1}{4}$	6	61	55	4	66	7–2	18,200	1900
131.9	22 2	10 0	$33\frac{1}{4}$	$50\frac{1}{4}$	$6\frac{1}{4}$	67	61	5	72	7–8	22,000	2100
167.5	22 2	10 6	$33\frac{1}{4}$	$52\frac{1}{4}$	$6\frac{1}{4}$	73	67	6	78	8–2	23,000	2500
197.1	22 2	11 0	$33\frac{1}{4}$	$54\frac{1}{4}$	$7\frac{1}{4}$	79	73	6	84	8–8	24,000	2625
242.0	22 2	11 6	$33\frac{1}{4}$	$54\frac{1}{4}$	$7\frac{1}{4}$	85	79	7	90	9–2	26,000	2850
283.0	22 2	12 0	$33\frac{1}{4}$	$56\frac{1}{4}$	$7\frac{1}{4}$	91	85	7	96	9–8	28,000	3050
319.7	22 2	12 6	$33\frac{1}{4}$	$56\frac{1}{4}$	$7\frac{1}{4}$	97	91	8	102	10–2	30,000	3400
354.6	24 2	12 6	$33\frac{1}{4}$	$56\frac{1}{4}$	$7\frac{1}{4}$	97	91	8	102	10–2	32,000	3400
391.6	24 2	12 6	$33\frac{1}{4}$	$56\frac{1}{4}$	$7\frac{1}{4}$	97	91	8	102	10–2	32,000	3400

Other size boilers are made between the sizes given by varying length from 14 to 22 ft. and by changing size and number of rows. The thickness of shell is given by the formula $pd = 2tS_x\,eff.$

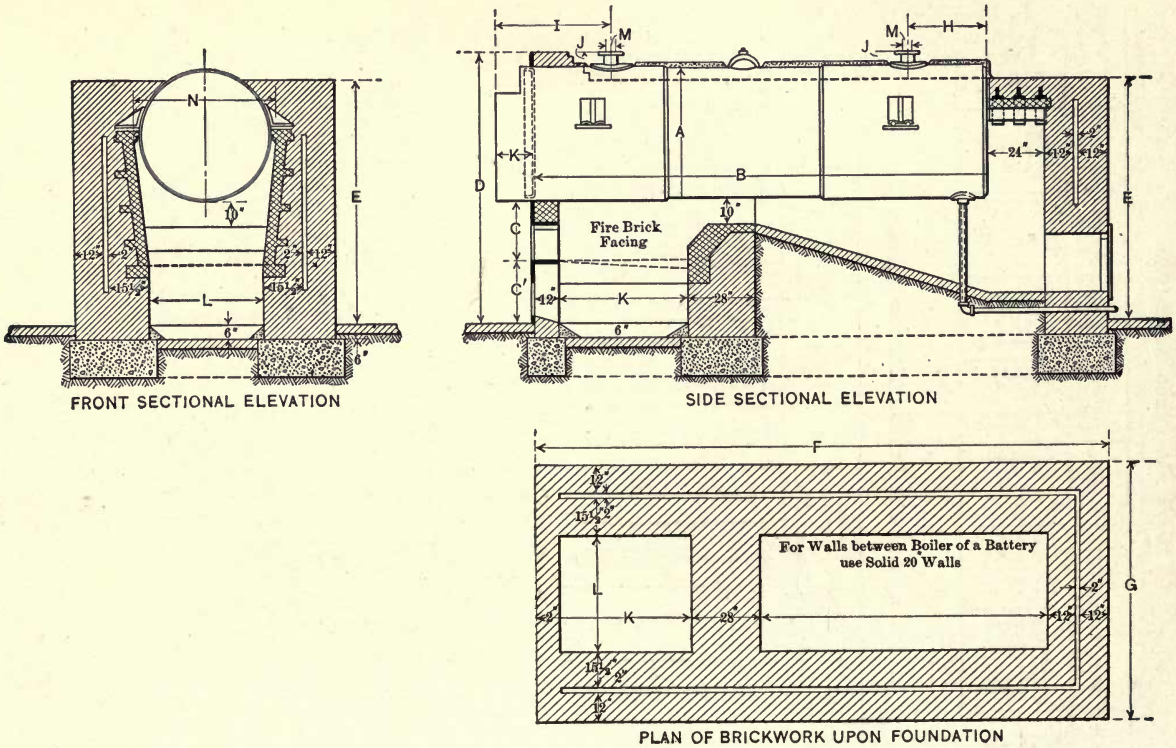


FIG. D.—Dillon Return Tubular Boilers.

WATER TUBE BOILER DIMENSIONS

Boiler H.P.....	75	100	200	300	400	500
Columns and Rows of 4" flues.....	5–9	6–9	10–9	16–9	16–12	21–12
A and number of drums.....	36" 1	36" 1	36" 2	42" 2	42" 2	42" 3
B.....	19' 0"	18' 4¾"	20' 4"	20' 4"	20' 4"	20' 4"
C.....	8' 2"	8' 2"	8' 2"	8' 2"	8' 2"	8' 2"
D.....	3' 2"	3' 2"	3' 2"	3' 2"	3' 2"	3' 2"
E.....	4½"	4½"	4½"	4½"	4½"	4½"
F.....	30"	39¾"	39¾"	39¾"	39¾"	39¾"
G.....	7' 5"	7' 5"	7' 2"	7' 2"	9' 2"	9' 2"
H.....	14' 8"	14' 8"	14' 8"	15' 2"	17' 6"	17' 6"
I.....	13' 0"	13' 0"	13' 0"	13' 3"	15' 7"	15' 7"
J.....	6' 0"	6' 0"	6' 0"	6' 0"	6' 0"	6' 0"
K.....	2' 3½"	2' 3½"	2' 3½"	2' 3½"	2' 3½"	2' 3½"
L.....	15"	15"	15"	15"	15"	15"
M.....	6' 4"	6' 4"	6' 4"	6' 4"	8' 2"	8' 2"
N.....	17' 9½"	17' 9½"	19' 9"	19' 9"	19' 9"	19' 9"
O.....	10' 9"	11' 11"	17' 2"	24' 2"	24' 2"	30' 0"
P.....	16"	16"	16"	16"	16"	16"
Q.....	6' 0"	6' 0"	7' 0"	7' 0"	7' 0"	7' 0"
R.....	24"	24"	24"	24"	24"	24"
S.....	8"	8"	8"	8"	8"	8"
T.....	17"	17"	17" & 24"	17" & 24"	17" & 24"	17" & 24"
U.....	3' 3"	3' 10"	6' 2"	9' 8"	9' 8"	12' 7"
W.....	9"	9"	12"	12"	15"	15"
X.....	15' 5"	15' 5"	15' 8"	16' 2"	17' 11"	18' 9"

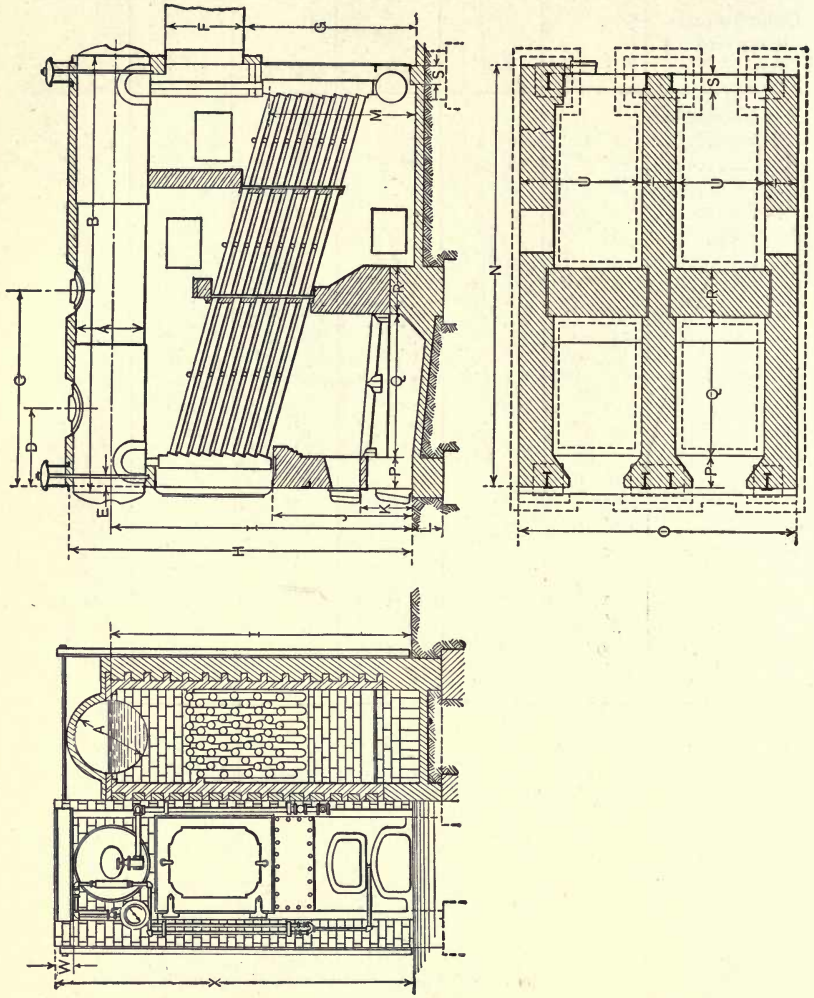


FIG. E.—Babcock and Wilcox Boiler.

ANTHRACITE SUCTION PRODUCERS. DIMENSIONS

H.P.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*K*	Width.	Height.
100	5′ 8″	11′ 8″	3′ 9″	3′ 3″	14′ 0″	6′ 6″	7′ 2″	5′ 4″	13′ 0″	20′ 0″
200	7′ 2″	12′ 8″	4′ 6″	4′ 8″	18′ 0″	7′ 0″	9′ 0″	6′ 3″	15′ 0″	21′ 0″
300	8′ 8″	15′ 3″	3′ 5″	5′ 8″	20′ 0″	8′ 9″	10′ 7″	6′ 8″	17′ 6″	26′ 6″
400	9′ 8″	16′ 0″	3′ 5″	6′ 6″	20′ 0″	9′ 3″	11′ 8″	7′ 1″	18′ 6″	26′ 6″

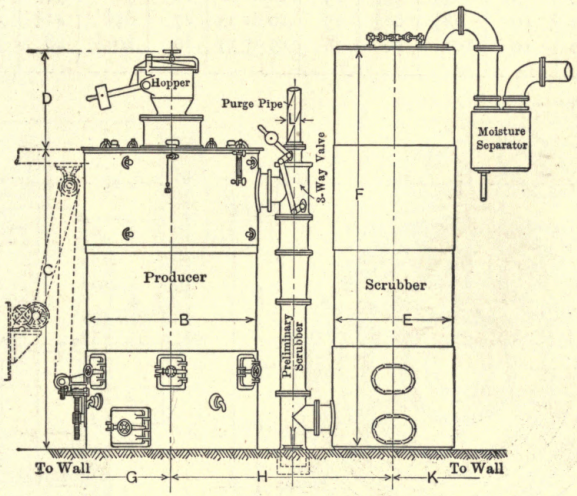


FIG. F.—Gas Producer.

DIRECT-CONNECTED GENERATOR

DIMENSIONS (Inches)

K.W.	Speed.	Poles.	*B*	*C*	*D*	*A*	*E*	*F*	*G*	*H*	*I*
25	280	6	$42\frac{7}{8}$	$34\frac{3}{8}$	$15\frac{1}{4}$	4 or $4\frac{1}{2}$	$18\frac{1}{4}$	39	$19\frac{1}{8}$	$18\frac{1}{4}$	$16\frac{3}{4}$
50	260	6	$48\frac{1}{4}$	$37\frac{1}{16}$	$15\frac{1}{4}$	$4\frac{1}{2}$ or $6\frac{1}{2}$	21	$44\frac{1}{2}$	$21\frac{3}{16}$	$19\frac{1}{2}$	19
75	250	6	$54\frac{7}{8}$	$43\frac{11}{16}$	19	$5\frac{1}{2}$ or $7\frac{1}{2}$	$22\frac{1}{4}$	51	$24\frac{11}{16}$	$21\frac{3}{4}$	21
100	235	6	$60\frac{1}{2}$	$46\frac{1}{2}$	19	6 or $8\frac{1}{2}$	$22\frac{1}{2}$	56	$27\frac{1}{2}$	24	$23\frac{3}{4}$
150	200	8	$68\frac{3}{4}$	$53\frac{5}{8}$	22	7 or 10	24	$64\frac{1}{2}$	$31\frac{5}{8}$	25	$27\frac{3}{4}$
200	180	8	$74\frac{3}{8}$	$59\frac{5}{8}$	25	8 or 11	26	71	$34\frac{5}{8}$	$29\frac{1}{2}$	29
300	150	10	$88\frac{1}{2}$	$66\frac{1}{4}$	25	10 or 13	27	$85\frac{1}{2}$	$41\frac{1}{4}$	$31\frac{1}{4}$	$31\frac{1}{4}$
400	150	10	101	76	28	15 or 17	30	$101\frac{1}{2}$	48	$34\frac{1}{2}$	35

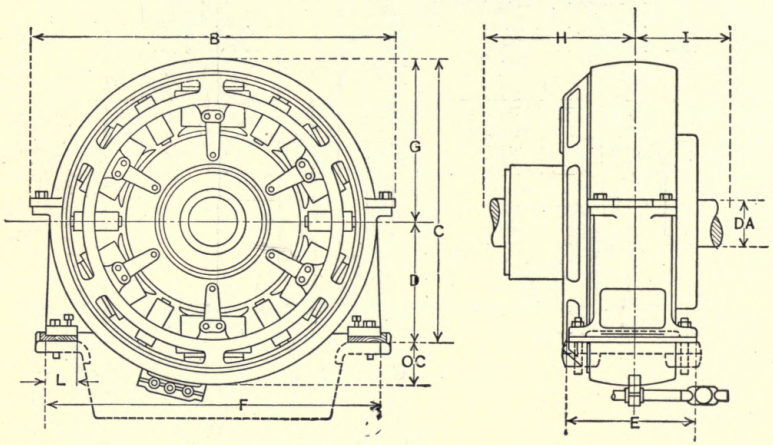


FIG. G.—Generator.

MOTOR DIMENSIONS (Inches)

H.P.	Speed.	Poles.	*A*	*B*	*C*	*D*	*E*	*F*	*G*
20	800	4	$52\frac{7}{16}$	$34\frac{1}{8}$	$26\frac{1}{2}$	$31\frac{3}{4}$	15	$25\frac{1}{16}$	28
50	650	6	$62\frac{1}{2}$	45	36	43	21	$26\frac{1}{4}$	$28\frac{1}{4}$
80	600	6	$75\frac{3}{4}$	52	$42\frac{3}{8}$	$49\frac{11}{16}$	24	31	$36\frac{3}{4}$
100	575	6	$90\frac{1}{8}$	$58\frac{3}{4}$	$46\frac{1}{8}$	$52\frac{1}{16}$	24	$35\frac{1}{8}$	$36\frac{3}{4}$
125	550	6	$93\frac{5}{8}$	62	$50\frac{1}{4}$	$57\frac{5}{8}$	27	$36\frac{5}{8}$	$37\frac{1}{2}$

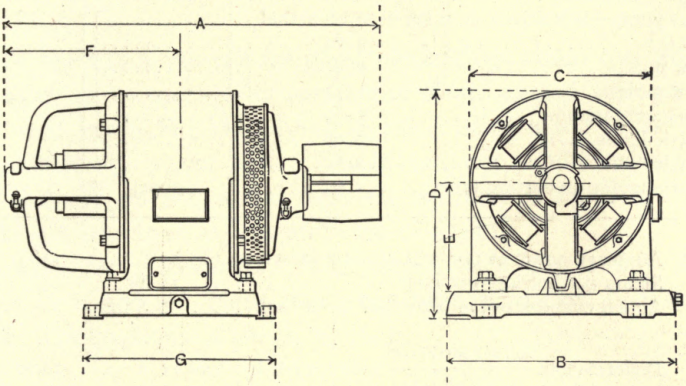


FIG. H.—Motor.

Power and Performance of Plants. *Auxiliary Power.* The power used by auxiliaries in a plant may be estimated from the following tables from which proportions may be found.

The power used in a 100-ton compression plate plant is given in the Transactions of the American Society of Refrigerating Engineers.

			By H. Torrance, Jr.	By T. Shipley.
1 H.P. Compressor			300.00	220
Water pump—belt driven 11.6 net—20.46 actual				
Agitator	4.3	6.60		
Thaw pump	0.3	0.53		
Boiler pump	0.98	1.71		30
Air vacuum pump		3.30		
Electric lights		10.00		
Electric crane		4.80		
Electric motor on cutting table		3.63	51.03	

Auxiliary power in per cent of main power..... 17% 15%
Steam per horse-power hour..... 18 lbs.
Actual evaporation in boiler per lb. of coal..... 8.8 lbs.

Performance: $\frac{8.5 \times 2000 \times 100}{351 \times 18 \times 24}$ tons ice per ton coal 11.2 10 to 15

The following data are taken from an electrically driven raw-water plant of 200 tons capacity:

Compressor motor	600 H.P.
1200-gallon cooling-tower pump motor	50
1600-gallon brine-pump motors	100
1200-cu.ft. air-compressor motor	30
Core-pump motor	5
Eight agitator motors	24
	209 H.P.

Auxiliary power in per cent of main power 35%.
Air used. $\frac{1}{2}$ to 1.8 cu.ft. free air per minute per 300-lb. can.

From a plant reported by W. H. Doreman, in *Ice and Refrigeration* for April, 1915, the following data are given:

Total horse-power per ton	3.7
Main motors	81.0%
Water pumps	7.2
Brine pumps	3.7
Air pump	1.8
Pressure air pump	2.7
Cranes	0.4
Agitators	4.2
	100.00

To operate a 150-ton ice plant electrically would cost about the same as to operate by compound steam engine with power at 1 cent per K.W., the saving in labor being made up by the increase in the power cost. The investment would be $20,000 less and 20% of this would represent an item of $4000 in favor of the electric drive at this unit cost of 1 cent.

Steam for Auxiliaries. The steam used per twenty-four hours in a 100-ton plate plant has been given by I. Warner in the Transactions of the A. S. R. E.

Compressor engine	125,703 lbs.	69.6%
Auxiliary engine and pump	45,347	25.2
Dynamo engine (12½ hrs.)	5,152	2.8
Harvesting 26 cakes	3,088	1.7
Melting off plates	1,290	0.7
	180,580 lbs.	100.0%

Performance: $\frac{8.5 \times 100 \times 2000}{180,580} = 9.4$ tons of ice per ton of coal.

Power and Performance of Absorption Plant. The power used in an absorption plant with plate ice is 58.33 I.H.P. per 100 tons capacity. The steam used is 27 lbs. per I.H.P. hour. The generator requires 50 lbs. of steam per hour per ton of capacity under usual conditions according to H. Torrance, Jr. The total steam needed by pumps is $58.33 \times 27 = 1575$ lbs. per hour. The total amount needed by generator is 5000 lbs. per hour. The exhaust from the pumps could be used in the gen-

erator. The radiation from the pumps may amount to one-third of the steam supplied or 525 lbs. of steam. This is lost. The remaining 1050 lbs. may be used, requiring 5525 lbs. per hour. The performance is

$$\frac{8.5 \times 2000 \times 100}{5525 \times 24} = 12.8 \text{ tons of ice per ton of coal.}$$

Torrance suggests operating an absorption plant with the exhaust from a compression plant. Using figures above and assuming 25% of the steam from the compression plant condensed by radiation, the steam returned would be

$$351 \times 18 \times 0.75 = 4740 \text{ lbs.}$$

The amount condensed would be

$$4740 \times \frac{0.25}{0.75} = 1580 \text{ lbs.}$$

The amount consumed would be

$$4740 + 1580 + 1575 = 7895.$$

The performance of the two machines together would be

$$\frac{8.5 \times 2000 \times 200}{7895 \times 24} = 18 \text{ tons of ice per ton of coal.}$$

Performance of Producer-driven Plant. A test of a producer plant for 144 hours reported in the transactions of the A. S. R. E. by E. W. Gallen Kamp, Jr., showed that 22.10 tons of ice were produced per ton of coal excluding auxiliaries or 17.8 tons with auxiliaries. A performance of 25 tons has been reported.

Labor Costs. The tables on pp. 373 and 374, arranged from averages of estimates given by a number of manufacturers, may be used to estimate the probable number of men and cost of labor:

Ice Plant—Steam Driven

Number of men and labor cost

Men.	Tons of Ice per Day (½ Ton of Refrigeration)									
	5	10	15	20	25	35	50	75	100	200
Engineer, day	1–2.50	1–2.50	1–2.50	1–3.00	1–3.00	1–3.50	1–4.00	1–4.50	1–5.00	1–6.00
Engineer, night	1–2.25	1–2.25	1–2.25	1–2.50	1–2.50	1–3.00	1–3.50	1–4.00	1–4.00	1–4.00
Tankman, day	1–1.75	1–2.00	1–2.00	1–2.00	1–2.00	1–2.00	1–2.00	2–4.00	2–4.00	4–8.00
Tankman, night	.....	.....	1–1.75	1–1.75	1–1.75	1–1.75	1–1.75	2–3.50	2–3.50	4–8.00
Fireman, day	.....	.....	.....	.....	1–2.00	1–2.00	1–2.00	1–2.00	1–2.00	2–4.00
Fireman, night	.....	.....	.....	.....	.....	1–2.00	1–2.00	1–2.00	1–2.00	1–2.00
Oiler, day	.....	.....	.....	.....	.....	1–1.75	1–1.75	1–2.00	1–2.00	1–2.00
Oiler, night	.....	.....	.....	.....	.....	.....	1–1.75	1–2.00	1–2.00	1–2.00
Laborer, day	.....	.....	.....	.....	.....	.....	1–1.75	1–1.75	2–3.50	3–5.25
Laborer, night	.....	.....	.....	.....	.....	.....	.....	1–1.75	2–3.50	2–3.50
Total	$6.50	$6.75	$8.50	$9.25	$11.25	$16.00	$20.50	$27.50	$31.50	$44.75
Labor cost per ton	1.30	0.67	0.57	0.46	0.45	0.46	0.41	0.36	0.32	0.22

Refrigerating Plant—Steam Driven

Number of men and labor cost

Men.	Tons of Refrigeration									
	5	10	15	20	25	35	50	75	100	200
Engineer, day	1–2.50	1–2.50	1–2.50	1–2.50	1–2.50	1–3.00	1–3.50	1–4.00	1–4.50	1–5.50
Engineer, night	1–2.25	1–2.25	1–2.25	1–2.25	1–2.25	1–2.50	1–3.00	1–3.50	1–4.00	1–4.00
Fireman, day	.....	.....	.....	.....	.....	1–2.00	1–2.00	1–2.00	1–2.00	1–2.00
Fireman, night	.....	.....	.....	.....	.....	1–1.75	1–1.75	1–2.00	1–2.00	1–2.00
Oiler, day	.....	.....	.....	.....	.....	.....	1–1.75	1–1.75	1–2.00	1–2.00
Oiler, night	.....	.....	.....	.....	.....	.....	.....	1–1.75	1–2.00	1–2.00
Laborer, day	.....	.....	.....	.....	.....	.....	.....	.....	.....	1–1.75
Laborer, night	.....	.....	.....	.....	.....	.....	.....	.....	.....	1–1.75
Total	$4.75	$4.75	$4.75	$4.75	$4.75	$9.25	$12.00	$15.00	$16.50	$21.00
Labor cost per ton	0.95	0.47	0.32	0.24	0.18	0.26	0.24	0.20	0.17	0.11

Note.—In the tables above omit firemen for oil-engine drive. For producer engine plants use the tables above. For electric motor-driven plants omit oilers and firemen from tables for costs of labor.

COST OF FUEL AND SUPPLIES PER DAY

	Tons of Ice per 24 Hours									
	5	10	15	20	25	35	50	75	100	200
Fuel cost at $3 per ton.....	3.25	6.50	9.50	12.00	13.00	16.00	24.00	34.00	43.00	78.00
Oil at 4 cts. per gal.....	3.20	4.20	4.80	4.40	5.00	7.00	10.00	14.00	20.00	40.00
Electricity at 2 cts. per K.W.	8.80	16.80	19.20	17.00	20.00	28.00	40.00	56.00	80.00	160.0
Oil, waste, etc.....	.50	.75	1.00	1.10	1.15	1.45	1.90	2.75	3.75	7.25

Cooling Water. Starr in *Ice and Refrigeration* for Sept., 1911, points out that the head pressure increases when the quantity of water is decreased. This increases the cost of compression but decreases the cost of water and the cost of pumping water. There may be some point at which the combined cost of compression, water and pumping water is a minimum. This point will vary with cost of water, lift of water and cost of compression. This should be investigated for any given problem.

Cost of Water. B. C. Sloat in *Ice and Refrigeration* for Dec., 1910, gives the following costs of pumping water per 1000 gallons:

Head lifted, feet.....	50	100	150	300
Deep-well pump.....	1.7 cts.	3.4	5.1	10.3
Air-lift pump.....	1.2	3.6	7.7	
Displacement pump.....	0.85 ct.	2.3	4.2	

Cities charge from 5 to 20 cts. per 1000 gallons for water.

Cost of Supplies in Ice Plant. Ice Plant of Moderate Size by Charles Dickerman in Transactions A. S. R. E., 1908.

Year.	Total Tons.	Tons per Day. 300 Days.	Cost Coal.	Wages.	Supplies.	Repairs.	Improvements.	General Expenses.
1904	6667	23	$1321.08	$4749.90	$1266.72	1354.40	$1352.02	
1905	8720	29	1550.16	4677.70	936.64	823.14	756.46	$ 679.23
1906	9144	30	1481.15	5398.55	837.25	1075.36	301.15	1614.48
1907	8866	30	1377.05	5204.19	887.51	933.80	556.00	604.26
Average per ton.....			$0.172	$0.600	$0.118	$0.125	$0.089	$0.087

Total cost per ton exclusive of overhead charges..... $1.19
Receipts per ton at plant..... $1.50 to $8

Data from plant:

Capacity, 30 tons.
Compressors, two 13×30 vertical.
Condensers, double pipe, 2 and 3-in. six banks, 12 high, 18 ft. tubes.
Brine tank, 24 coils 2-in. pipe, 6 high, 44 ft. long, 6336 ft.
Ice cans, 483 cans, 300 lbs.
Two bulkheads and two engine-driven agitators in tank.
Fore cooler 12-in. diam.×16 ft. long.
Ice house, 60 tons capacity, cooled by brine.
Boiler, 66-in. return tubular, 18 ft. long. Fortyeight 4½-in. long.
Stack, 30-in., 125 ft. high.
Bituminous coal, $2.40 to $2.50 per ton.
Water, 3 cts. per ton of ice.
Ammonia, $150 to $200 per year.
Performance, 6½ tons of ice per ton coal.

Cost of labor, fuel and supplies at a plant in Asbury Park was 85 cts. per ton, one-half of which was labor cost.

Load Factors. The load factor sometimes assumed covers one-third year at full capacity, one-third year at half capacity and one-third at quarter capacity. This gives

$$\tfrac{1}{3}\times 1 = .333$$
$$\tfrac{1}{3}\times \tfrac{1}{2} = .167$$
$$\tfrac{1}{3}\times \tfrac{1}{4} = .083$$

$$\text{Load factor} \quad 0.583 = 58\%$$

Nordmeyer suggests that the operation at full capacity for July with no storage capacity represents 15% of year's demand (use of ice in July equals 15% of total yearly amount)

$$\text{Load factor} = \frac{1}{\frac{.15}{12}} = 0.56 = 56\%.$$

With storage space the plant may be run at full capacity for even the whole year. Of course the cost of storage is offset partially or completely by the smaller fixed charges on the smaller equipment.

Cost of Storage. W. E. Parsons states that it cost 25 cents per ton to store ice, hold it from spring until midsummer and remove it to delivery platform in a 75-ton plant. J. N. Briggs increases this to 45 cents per ton to cover the charge for the storehouse.

Initial and Operating Costs of Small Refrigerating Plants

Robert P. Kehoe, *Power*, May, 1915.

	Capacity in Tons of Refrigeration per 24 Hours.											
	10 Tons.			15 Tons.			20 Tons.			25 Tons.		
	Steam.	Electric.	Oil.	Steam.	Electric.	Oil.	Steam.	Electric.	Oil.	Steam.	Electric.	Oil.
Investment mechanical equipment	$5000	$4500	$5300	$7000	$6400	$7400	$8000	$7300	$8400	$9200	$8400	$9500
Cost of labor for 24 hours if men can be used elsewhere	$2.00	$1.50	$1.50	$3.00	$2.00	$2.00	$3.50	$2.50	$2.50	$4.00	$3.00	$3.00
Cost of fuel or power per 24 hours: coal at $3.50, oil at 3½ cts., current at 2 cts.	3.50	4.80	1.50	4.75	7.20	1.80	6.00	9.60	2.20	7.00	12.00	2.50
Supplies per 24 hours, NH_3, oil, waste, etc.	0.75	0.75	0.75	1.00	1.00	1.00	1.25	1.25	1.25	1.50	1.50	1.50
Net operating expenses	$6.25	$7.05	$3.75	$8.75	$10.20	$4.80	$10.75	$13.35	$5.95	$12.50	$16.50	$7.00
Total expense per ton	$1.00	$1.00	$0.72	$0.93	$0.96	$0.64	$0.84	$0.92	$0.50	$0.78	$0.89	$0.54

	Capacity in Tons of Ice per 24 Hours											
	10 Tons.			15 Tons.			20 Tons.			25 Tons.		
	Steam.	Electric.	Oil.	Steam.	Electric.	Oil.	Steam.	Electric.	Oil.	Steam.	Electric.	Oil.
Investment mechanical equipment	$8500	$7500	$8500	$11,500	$10,000	$11,500	$12,500	$11,000	$12,500	$15,000	$13,300	$15,000
Buildings	3,500	3,000	3,000	4,500	4,000	4,000	5,500	5,000	5,000	6,500	6,000	6,000
Total investment exclusive of land.	$12,000	$10,500	$11,500	$16,000	$14,000	$15,500	$18,000	$16,000	$17,500	$21,500	$19,300	$21,000
Operating expenses: Labor 24 hours	$7.50	$7.50	$7.50	$9.50	$9.50	$9.50	$10.00	$10.00	$10.00	$10.50	$10.50	$10.50
Cost of fuel or energy at above rates	7.00	12.00	2.63	10.00	18.00	3.94	13.00	23.00	5.27	15.50	30.00	6.56
Supplies as above	1.50	1.50	1.50	2.25	2.25	2.25	3.00	3.00	3.00	3.75	3.75	3.75
Total operating expenses	$16.00	$21.00	$11.63	$21.75	$29.75	$15.69	$26.00	$36.00	$18.27	$29.75	$44.25	$20.81
Cost per ton at 58% load factor and 10% on investment	$3.08	$3.45	$2.61	$2.72	$3.16	$2.33	$2.38	$2.83	$1.97	$2.31	$2.69	$1.82

Costs of 100-ton Ice Plant

By W. T. Price, in *Ice and Refrigeration* for Dec., 1915.

INVESTMENT

	Steam Plant.	Oil Engine Direct.	Oil Engine Belted.	Producer Gas.
Engine and compressor	$27,000	$ 34,000	$ 38,000	$ 39,000
Freezing system	27,000	27,000	27,000	27,000
Ice-storage piping	1,500	1,500	1,500	1,500
Building	37,000	29,000	31,000	35,000
Land	9,500	8,700	9,100	10,000
	$102,000	$100,200	$106,600	$112,500

OPERATING COSTS

Coal $3.51 per ton with 8 : 1 boiler-evaporation	$9,450			
Fuel oil 3½ cts. per gal. 1 ton to 4 gal	.....	$3,020		
Fuel oil 3½ cts. per gal., 1 ton to 4½ gal	.....	.....	$3,400	
Pea coal, anthracite, $4.50 per ton, 26 tons ice per ton	.....	.....	.....	$3,740
Supplies: oil ammonia, waste, 6 cts. per ton with steam, 7 cts. per ton with oil and gas	1,300	1,500	1,500	1,500
Labor 365 days	10,600	10,800	10,800	12,250
Repairs, 3%	3,060	3,000	3,170	3,470
Depreciation:				
Buildings 3%	1,110	870	930	1,050
Equipment 5%	2,770	3,120	3,320	3,370
	$28,290	$22,310	$23,120	$25,380
Assuming 33% full capacity, 33% ½ capacity, 33% ¼ capacity gives 21,600 tons at	$1.31	$1.03	$1.07	$1.17

DAILY LABOR COST

Engineer chief	$5.00	$6.00	$6.00	$6.00
Engineer assistant	3.50	3.50	3.50	3.50
Oilers, 2 shifts	4.00	4.00	4.00	4.00
Firemen, 2 shifts	4.50	.....	....	4.00
Tankmen	8.00	12.00	12.00	12.00
Storehouse men	4.00	4.00	4.00	4.00
Total	$29.00	$29.50	$29.50	$33.50

Cost of Ice Plants for Steam and Oil Engine Operation

L. K. Doelling, A. S. R. E. Journal, Sept., 1915.

INVESTMENT COSTS

Apparatus.	Steam Engine Plant.			Oil Engine Plant.		
	50 tons.	100 tons.	200 tons.	50 tons.	100 tons.	200 tons.
Oil Engine, rope drive and foundation	.....	.....	.....	$14,500	$25,500	$45,000
Piping for oil, water exhaust and oil tank	.....	.....	.....	1,500	2,500	4,500
Boiler plant and foundation	$3,000	$5,500	$10,000			
Piping for steam, exhaust and water	1,500	2,000	3,000			
Steam engine and foundation direct connected	5,000	9,500	15,500			
Compressor, condenser and ammonia piping	5,500	10,000	18,000	5,500	10,000	18,000
Freezing system	14,000	27,000	50,000	14,000	27,000	50,000
Ice Storage	1,000	1,500	2,500	1,000	1,500	2,500
Buildings	16,000	35,000	60,000	16,000	35,000	60,000
Land	5,000	8,000	12,000	5,000	8,000	12,000
Total	$51,000	$98,500	$171,000	$57,500	$109,500	$192,000

OPERATING COSTS

Labor, fuel and ammonia for for 216 days full capacity	$10,314	$16,740	$29,268	$6,454	$10,908	$18,360
Labor for remainder of year	2,736	4,174	5,832	2,348	3,072	5,328
5% depreciation on equipment	1,500	2,775	4,950	1,825	3,325	6,000
3% depreciation on building	480	1,050	1,800	480	1,050	1,800
5% on total investment for taxes, repairs, water and incidentals (no allowance for interest on investment)	2,550	4,925	8,550	2,875	5,475	9,600
Total	$17,580	$29,664	$50,400	$13,982	$24,430	$41,088
Tons per year	10,800	21,600	43,000	10,800	21,600	43,000
Cost per ton	$1.62	$1.40	$1.17	$1.30	$1.11	$0.96
(Cost with interest)	(1.86)	(1.63)	(1.36)	(1.57)	(1.35)	(1.13)

DAILY OPERATING EXPENSE

Labor	$19.00	$29.00	$40.50	$16.00	$25.50	$37.00
Fuel (coal $3.50, oil at 3.5 cts)	22.75	38.50	77.00	7.88	15.00	30.00
Ammonia, oil, waste	6.00	10.00	18.00	6.00	10.00	18.00
Total	$47.75	$77.50	$135.50	$29.88	$50.50	$85.00

H. Swan shows that although a compound-engine plant would cost 12% more than the steam-engine plant above the fuel cost would be so much reduced that the cost of ice would be reduced by 10%.

L. C. Nordmeyer gives the following costs for a 100-ton plant at 57% load factor.

COST OF PLANT

	Simple Steam Engine.	Compound Condensing Steam Engine.	Diesel Oil Engine.
Buildings	$60,000	$60,000	$60,000
Machinery	65,000	76,400	83,923
	$125,000	$136,400	$143,923

OPERATING COSTS PER TON PRODUCED

Cost of water per ton	.....	.....	$0.09
Cost of fuel per ton	$0.976	$0.753	0.226
At 25 cts. per 42 gals.	(1.03 bbl. per ton)	(0.79 bbl. per ton)	(0.238 bbl. per ton)
Fixed charges per ton on machinery	0.546	0.642	0.821
Operating cost per ton	0.579	0.579	0.516
Total cost per ton	$2.101	$1.974	$1.653
Fixed charges per ton for building	0.504	0.504	0.504
Total	$2.605	$2.478	$2.157

	Equipment: Two 60-ton refrigerating machines and engines. Water tube boiler and feed pumps. Feed water heater, chimney. Freezing system. Steam condensers. Ammonia condensers. Air lift pumps, air compressors. Circulating water pump. Cooling tower. Brine pump and cooler. Piping for storage. Piping for apparatus and covering. 60 K.W. generator and engine. Ammonia, calcium chloride. Foundations.	Equipment: Two 60-ton compressors belted to two 225 H.P. Diesel engines. Two 40 K.W. generators. Raw water freezing system. Cooling tower. Two circulating pumps. Two brine pumps. Cold storage piping. Piping and covering. Ammonia, calcium chloride. Two-oil tanks. Foundations. Ammonia condenser.

In the above tables allowance has not always been made for interest on the total investment. The items should be carefully gone over and the following percentages allowed:

Interest on total cost (return on investment)	5 to 8%
Taxes	1 to 2
Insurance ($\frac{1}{4}$% fire proof, 1% frame)	$\frac{1}{4}$ to 1
Depreciation on machinery (about 14 years life at 5%)	$4\frac{1}{2}$
Depreciation on buildings (25 years life at 5%)	2
Repairs to machinery	3 to 5
Repairs to building	1 to 2
Total	$16\frac{3}{4}$ to $24\frac{1}{2}$

Ice Delivery Data:

Cost of wagons	$275.00
Cost of horses	300.00
Cost of harness	35.00
Cost of feed	.50 per day
Cost of drivers	2.75 "
Cost of helpers	2.00 "

Mr. G. T. Lawrence gives the following in *Ice and Refrigeration*, Apr., 1913.

Cost of 1 team 1 horse per month.

Feed, shoeing, driver, hostler, repairs	$81.12
Insurance	1.68
Depreciation, $8\frac{1}{3}\%$ on $720	5.00
Interest, 5% on $720	3.00
	$90.80

Cost of 3-ton truck per month:

Driver, repairs, oil, gasoline and tires	$138.00
Insurance	10.00
Depreciation, $16\frac{2}{3}\%$ on $3600 per month	50.00
Interest, 5% on $3600 per month	15.00
	$213.00

One truck can do the work of 2.93 teams costing $266.04 per month.

Amount retail delivered per day per man . . . . . $2\frac{1}{2}$ tons

In *Ice and Refrigeration* for Dec., 1914, the following data are given for 5-ton truck:

Cost per day	$18.45
Cost per mile	0.39
Cost per hour	3.80

Cost computed on equal time standing and running.

Refrigerated Railroad Car Data

Cost of refrigerator car		$1600
Cost of box car		1100
Weight of refrigerator car		46,800 lbs.
Capacity		15,000 to 35,000 lbs.
Length:	Over couplers	44′ $3\frac{1}{2}$″
	Over sheathing	40′ $11\frac{1}{8}$″
	Inside	39′ $10\frac{3}{4}$″
	Inside between ice tanks	33′ $2\frac{1}{4}$″
Width:	Over sheathing	9′ $2\frac{5}{8}$″
	Inside	8′ $2\frac{3}{4}$″
	Gauge track	4′ $8\frac{1}{2}$″
Height:	Rail to top of brake shaft	13′ $5\frac{13}{16}$″
	Rail to running board	13′ $\frac{1}{2}$″
	Rail to eaves	12′ $3\frac{11}{16}$″
	Rail to coupler	2′ $10\frac{1}{2}$″
	Inside	7′ $5\frac{3}{16}$″
Cubic capacity:	Total	2444 cu.ft.
	Available	2032 cu.ft.
Ice capacity at each end		11,000 lbs.

Insulation, roof: $\frac{1}{2}$″ 3-ply flax felt
$\frac{3}{8}$″ ship lap, 2 ply
$\frac{5}{8}$″ ceiling

Sides: $\frac{13}{16}$″ siding, $\frac{5}{8}$″ furring, 2 ply
$\frac{3}{8}$″ shiplap
$\frac{1}{2}$″, 2 ply, flax felt
$\frac{13}{16}$″ lining

Floor: $\frac{1}{2}$″ flax felt, 3 ply
$1\frac{3}{4}$″ ship lap floor
$\frac{3}{8}$″ ship lap, 3 ply

Average amount of ice put in at each station	5345 lbs. per car
Cost of icing, Chicago to New York	$16.00
Cost of icing, California to Chicago	62.50
Cost of ice and salt per ton at Indianapolis	2.85
Cost of cleaning cars	31 to 80 cents
Cost of stripping cars	$2.20

Rates of Precooling by Mr. A. Faget:

Asparagus cars 25° per hour air heated in passage from 16° to 38°.

Celery cars, 16° per hour air heated in passage from 14° to 40°

Grape cars, 14° " " " " 14° to 31°

Orange cars, 8° " " " " 10° to 30°

Air employed, 8000 cu.ft. per car per minute.

Charge for precooling	$25.00	per car
Charge for precooling and first ice	55.00	"
Charge for icing to Chicago	62.50	"
Charge for icing to New York	75.00	"
Cost of precooling and icing	32.50	"
Charge for use of car	7.50	"

Ice for Passenger Cars

20 lbs. of ice per car per 300 miles.

Rink Data

Use $1\frac{1}{4}$-in. brine pipe, 4-in. centers; using about 2 to 3 lin.ft. of pipe per square foot of surface. This may be formed in metal pan placed on 3-in. cork boards and fed from a brine main.

Ice Cream Data

(W. W. Wren, *Ice and Refrigeration*, May, 1915):

Cost per gallon:	Milk and cream	28.4 cts.
	Sugar	3.9
	Ice	5.9
	Salt	2.0
	Fruit	1.1
		41.3 cts.
Over-run or swell		$68\frac{1}{2}\%$
Shrinkage		6.1%

42 to 45 lbs. of ice per gallon of ice cream.

Temperature for hardening (6 to 8 hrs.) 0° F.

Temperature for ice making..... 15°
Temperature for ripening (12 hrs.) .. 33°
Mixing tank to be refrigerated with brine-freezing machine 15 minutes to batch.
Power: 1 H.P. for 5-gallon freezer, single.
1 H.P. for 50 gallons per hour in gangs from 1 motor.

Mr. J. H. Stone in the Transactions A. S. R. E. for 1910, gives the following data for warehouses using insulation of various values when the temperature of storage is 30° and the average outside temperature for eight months is 70°.

B.t.u. per Sq. ft. per 24 hrs. per Degree.	Cost of Insulation per Sq. ft.	Cost of Insulating 10000 Sq. ft.	Tons of Refrigeration.	Total Tons per Season.	Increase Cost of Insulation.	Decrease Cost of Machinery at $450 per Ton.	Net Decrease in Total Cost.	Increase in Fixed Charges at 15%.	Decrease in Operation at $1.00 per Ton.	Net Decrease in Cost, Dollars.
4	18 cts.	$1800	5.6	1333						
3	22	2200	4.2	1000	$400	$630	$230	−$34.50	$333	$367.50
2	27	2700	2.8	667	500	630	130	− 19.50	333	352.50
1.5	36	3600	2.1	500	900	315	− 585	87.75	167	79.25
1	50	5000	1.4	332	1400	315	−1085	212.75	168	−44.75

Note: The last three columns are changed from those given by Stone. This table gives decreased cost of any condition on one line over that of the preceding line. If money invested is the important item the 2 B.t.u. condition is the best; if operation, then 1.5 B.t.u. is the best condition. In this analysis the value of space taken by insulation is not considered.

STUDY FOR MINNEAPOLIS FOR COST OF NATURAL ICE AND MANUFACTURED ICE FOR 100,000 TONS

Ice and Refrigeration, March, 1915

INVESTMENT

Natural ice, storehouse for 100,000 tons.....	$125,000
Manufactured ice—Plant (275 tons).....	400,000
Storage 60,000 tons.....	60,000
Land.....	20,000
	$480,000

Cost of Production per Ton

Natural Ice

Delivery to storage	$0.30
Loss	0.04
Delivery to platform	0.15
6% interest on $1.25 per ton	0.075
10% depreciation on $1.25 per ton	0.125
2% taxes	0.025
3% insurance	0.035
3½% sinking fund	0.042
	$0.792
Freight	0.400
10% shrinkage in cars	0.132
Cars to wagon	0.25
	$1.574
Delivery and shrinkage	2.750
Overhead charges 10%	0.432
Selling price	$4.756

Manufactured Ice

Manufacturing cost	$1.00
6% interest on $4.80	0.288
1% insurance on $4.80	0.049
Repairs	0.192
2% taxes	0.096
3½% sinking fund	0.160
	$1.785
Delivery and shrinkage	2.642
Overhead charges 10%	.442
Selling price	$4.869

Data from Study of Household Refrigerators in Rochester, N. Y.

By John R. Williams

Weekly Amounts Ice.		Cost of Ice per Year.		Temperatures.	
50 lbs. or less	7%	Under $5	23%	In refrigerators:	
51 to 75 lbs.	12%	$5 to $10	43%	Below 45°	14%
76 to 100 lbs.	18%	$10 to $15	15%	45 to 50°	27%
101 to 200 lbs.	47%	$15 to $20	7%	50 to 60°	51%
201 to 300 lbs.	10%	$20 and over	12%	Over 60°	8%
301 and over	6%		100%	Living Rooms:	
	100%			Below 60°	0%
				60 to 70°	42%
				Above 70°	58%
				Cellars:	
				Below 55°	0%
				Below 60°	8%
				Above 60°	92%

Properties of Saturated Ammonia, NH_3

With Permission of G. A. Goodenough

Temperature Deg. F.	Pressure, Lbs. per Sq.in.	Heat Content of Liquid, B.t.u. per Lb.	Heat of Vaporization, B.t.u. per Lb.	Heat Content of Vapor, B.t.u. per Lb.	Internal Latent Heat, B.t.u. per Lb.	External Work, B.t.u. per Lb.	Entropy of Liquid.	Entropy of Vaporization.	Entropy of Vapor.	Specific Volume of Liquid, Cu.ft. Per Lb.	Specific Volume of Vapor, Cu.ft. per Lb.	Specific Weight, Lb. per Cu.ft.	Temperature.
t	p	i'	r	i''	ρ	ψ	s'	r/T	s''	v'	v''	m	t
−25	15.6	−59.8	591.1	531.3	542.1	49.0	−0.129	1.360	1.231	0.024	16.95	0.059	−25
−20	17.9	−54.6	587.4	532.8	538.0	49.4	−0.117	1.336	1.219	0.024	14.89	0.067	−20
−15	20.5	−49.4	583.6	534.3	533.9	49.7	−0.105	1.317	1.207	0.024	13.15	0.076	−15
−10	23.3	−44.2	579.9	535.7	529.8	50.1	−0.094	1.290	1.196	0.024	11.63	0.086	−10
− 5	26.5	−38.9	576.1	537.1	525.6	50.5	−0.082	1.267	1.185	0.024	10.32	0.097	−5
0	29.9	−33.7	572.2	538.5	521.4	50.8	−0.071	1.245	1.174	0.024	9.19	0.109	0
5	33.8	−28.4	568.3	539.9	517.1	51.2	−0.059	1.223	1.164	0.024	8.20	0.122	5
10	38.0	−23.2	564.4	541.2	512.9	51.5	−0.048	1.202	1.153	0.025	7.34	0.136	10
15	42.7	−17.9	560.4	542.5	508.6	51.8	−0.037	1.181	1.143	0.025	6.58	0.152	15
20	47.7	−12.6	556.3	543.7	504.2	52.1	−0.026	1.160	1.134	0.025	5.92	0.169	20
25	53.3	− 7.3	552.2	545.0	499.8	52.4	−0.015	1.140	1.124	0.025	5.34	0.187	25
30	59.4	− 1.9	548.1	546.2	495.4	52.7	−0.004	1.119	1.115	0.025	4.82	0.208	30
35	65.9	3.5	543.9	547.4	491.0	52.9	0.006	1.100	1.106	0.025	4.36	0.229	35
40	73.0	8.9	539.7	548.5	486.5	53.2	0.017	1.080	1.097	0.025	3.96	0.253	40
45	80.8	14.3	535.3	549.7	481.9	53.4	0.028	1.061	1.089	0.026	3.60	0.278	45
50	89.1	19.8	531.0	550.8	477.3	53.7	0.039	1.042	1.081	0.026	3.28	0.305	50
55	98.0	25.3	526.5	551.0	472.7	53.8	0.049	1.023	1.072	0.026	2.99	0.334	55
60	107.7	30.9	522.0	552.9	468.0	54.0	0.060	1.005	1.065	0.026	2.73	0.366	60
65	118.1	36.5	517.5	554.0	463.3	54.2	0.071	0.986	1.057	0.026	2.50	0.400	65
70	129.2	42.1	512.8	555.0	458.5	54.3	0.081	0.968	1.050	0.026	2.30	0.435	70
75	141.1	47.8	508.1	556.0	453.7	54.4	0.092	0.950	1.042	0.027	2.11	0.474	75
80	153.9	53.6	503.4	557.0	448.8	54.6	0.102	0.933	1.035	0.027	1.94	0.516	80
85	167.4	59.4	498.5	557.9	443.9	54.6	0.113	0.915	1.028	0.027	1.79	0.559	85
90	181.8	65.3	493.5	558.9	438.9	54.6	0.124	0.898	1.022	0.027	1.65	0.606	90
95	197.3	71.3	488.5	559.8	433.9	54.6	0.134	.881	1.015	0.027	1.52	0.656	95
100	213.8	77.3	483.4	560.7	428.7	54.7	0.145	0.864	1.009	0.028	1.41	0.710	100
105	231.2	83.4	478.2	561.6	423.5	54.7	0.156	0.847	1.003	0.028	1.30	0.766	105
110	249.6	89.6	472.9	562.5	418.3	54.6	0.166	0.830	0.997	0.028	1.21	0.826	110
115	269.2	95.9	467.4	563.3	412.9	54.5	0.177	0.814	0.991	0.028	1.12	0.891	115
120	289.9	102.2	461.9	564.2	407.5	54.4	0.188	0.797	0.985	0.028	1.04	0.960	120
125	311.6	108.7	456.3	565.0	402.0	54.3	0.199	0.781	0.979	0.029	0.97	1.031	125

Properties of Superheated Ammonia, NH_3

Arranged from Goodenough

Pressure, Lbs. per Sq. in.	Saturation Temperature.	Entropy																					
		I.11			I.12			I.13			I.14			I.15			I.16			I.17			
		Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.
15	−26.4																					
17	−21.9																					
19	−17.8																					
21	−14.0																					
23	−12.2																					
25	−7.2																					
28	−2.7																					
30	−0.1																					
34	5.3	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	11	543.5	8.28
38	10 0	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	15	544.2	7.45	23	549.9	7.63
42	14.4	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	19	545.2	6.79	27	549.9	6.96	34	554.4	7.07
46	18.4	.....	.....	.....	.....	.....	.....	.....	.....	.....	21	544.8	6.17	28	549.6	6.30	37	554.6	6.47	45	559.7	6.66
50	22.1	.....	.....	.....	.....	.....	.....	22	544.3	5.67	30	549.4	5.80	38	554.0	5.91	46	559.0	6.04	55	564.4	6.18
55	26.4	.....	.....	.....	.....	.....	.....	33	550.0	5.28	41	554.5	5.39	49	559.6	5.51	57	564.6	5.62	67	569.9	5.78
60	30.5	.....	.....	.....	35	549.8	4.84	43	554.1	4.93	51	559.3	5.04	59	564.4	5.15	68	569.8	5.27	78	575.3	5.38
65	34.3	.....	.....	.....	44	553.0	4.55	52	558.9	4.64	60	563.6	4.74	69	569.1	4.85	78	574.5	4.95	88	580.8	5.05
75	41.3	50	554.8	3.96	60	561.5	4.07	68	566.9	4.17	77	571.9	4.25	86	577.4	4.34	96	582.6	4.43	105	588.5	4.52
85	47.6	68	563.6	3.63	76	568.7	3.70	84	574.0	3.78	93	579.2	3.86	102	585.0	3.94	112	590.9	4.02	122	596.8	4.15
95	53.3	82	570.0	3.32	89	575.7	3.38	98	580.4	3.46	107	586.3	3.53	117	592.8	3.60	127	598.2	3.68	137	604.1	3.77
105	58.6	92	576.1	3.08	102	581.6	3.14	111	586.4	3.20	120	592.5	3.27	130	598.5	3.34	140	604.4	3.41	150	610.2	3.48
115	63.6	104	581.5	2.87	104	587.0	2.92	123	592.4	2.99	132	598.2	3.05	142	604.2	3.11	152	610.3	3.17	163	616.7	3.24
125	68.1	114	586.8	2.69	122	592.6	2.74	134	598.0	2.79	144	604.1	2.85	154	610.2	2.91	164	616.1	2.97	175	622.5	3.04
140	74.5	130	593.2	2.45	140	599.5	2.50	149	605.3	2.55	159	611.3	2.60	170	617.9	2.67	181	624.4	2.73	192	630.2	2.78
155	80.4	143	600.0	2.26	151	606.3	2.31	163	612.0	2.36	173	618.3	2.41	184	624.8	2.46	195	631.3	2.51	207	637.9	2.56
170	85.9	156	606.0	2.11	166	615.3	2.17	176	618.4	2.19	186	624.8	2.24	197	631.5	2.28	209	637.9	2.33	222	645.0	2.37
190	92.7	172	613.6	1.92	182	619.8	1.96	192	626.3	2.00	203	632.9	2.04	214	639.6	2.09	225	646.5	2.13	238	653.6	2.18
210	98.9	186	620.5	1.77	196	627.9	1.81	207	633.2	1.85	218	640.0	1.88	230	647.2	1.92	241	653.6	1.96	254	660.7	2.00
230	104.7	200	627.1	1.65	210	633.0	1.68	221	640.3	1.71	232	646.1	1.75	244	654.0	1.79	256	661.0	1.83	268	668.3	1.87
250	110.1	211	632.5	1.53	222	639.3	1.57	233	646.0	1.60	245	653.0	1.64	257	660.4	1.67	269	667.5	1.71	276	675.0	1.74

PROPERTIES OF SUPERHEATED AMMONIA—*Continued*

Arranged from Goodenough Tables

Pressure, Lbs. per Sq. in.	Saturation Temperature, Deg. F.	ENTROPY																					
		I.18			I.19			I.20			I.21			I.22			I.23			I.24			
		Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	
15	−26.4	.....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	−22	533.2	17.80	
17	−21.9	.....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	−17	535.2	15.87	−8	539.8	16.27	
19	−17.8	.....	.....	....	....	.....	.....	.....	.....	.....	.....	.....	.....	−13	536.3	14.28	−5	540.7	14.60	+3	545.1	14.93	
21	−14.0	.....	.....	....	....	.....	.....	.....	.....	.....	−10	536.9	13.00	−2	541.6	13.25	7	546.3	13.57	15	550.9	13.85	
23	−12.2	.....	.....	....	....	.....	.....	−8	537.1	11.90	0	541.7	12.10	0	546.4	12.40	17	551.1	12.70	25	555.8	12.96	
25	−7.2	.....	.....	....	....	.....	.....	1	541.4	11.10	9	546.0	11.40	18	550.6	11.60	26	555.4	11.90	35	560.1	12.14	
28	−2.7	.....	.....	....	5	542.4	10.1	13	547.2	10.30	22	552.0	10.50	30	556.6	10.70	39	561.6	10.90	48	566.6	11.14	
30	−0.1	5	541.1	9.31	12	545.9	9.57	21	550.9	9.72	29	555.4	9.92	38	560.4	10.13	47	565.5	10.34	57	570.9	10.57	
34	5.3	18	547.8	8.46	27	552.6	8.68	36	559.1	8.83	45	563.1	8.98	54	568.5	9.24	63	572.1	9.42	73	578.6	9.62	
38	10.0	31	553.9	7.78	39	558.5	7.94	48	563.6	8.11	57	568.8	8.28	67	574.4	8.46	77	579.8	8.65	87	585.2	8.83	
42	14.4	43	559.5	7.22	52	564.7	7.37	61	570.1	7.53	70	575.1	7.68	80	580.6	7.85	90	586.0	8.02	100	591.5	8.18	
46	18.4	54	564.6	6.75	63	570.0	6.88	72	575.3	7.02	81	580.2	7.15	91	585.8	7.30	101	591.3	7.45	112	597.3	7.62	
50	22.1	64	569.4	6.33	73	574.9	6.45	82	580.0	6.58	92	585.6	6.72	102	591.1	6.86	112	596.6	7.00	124	602.7	7.15	
55	26.4	75	575.0	5.91	84	580.4	6.05	94	585.8	6.16	104	591.4	6.28	115	597.2	6.40	125	602.7	6.51	136	608.8	6.64	
60	30.5	86	580.6	5.48	96	586.0	5.64	106	591.6	5.74	116	597.2	5.83	127	603.3	5.97	137	608.8	6.08	148	614.7	6.20	
65	34.3	97	586.2	5.15	106	591.7	5.26	116	597.5	5.37	126	603.1	5.49	137	609.3	5.61	149	614.8	5.72	160	620.6	5.84	
75	41.3	115	594.2	4.62	125	599.9	4.72	135	605.7	4.81	146	612.0	4.91	157	618.2	5.00	169	624.2	5.12	180	630.3	5.22	
85	47.6	132	602.6	4.19	143	608.3	4.27	155	614.6	4.37	166	620.7	4.54	176	626.5	4.55	189	636.2	4.64	200	640.2	4.74	
95	53.3	147	609.8	3.84	158	615.9	3.92	169	622.1	4.00	180	628.3	4.08	192	634.9	4.17	204	642.0	4.26	216	648.1	4.34	
105	58.6	160	616.6	3.55	171	622.9	3.63	182	629.2	3.71	194	635.7	3.78	206	642.6	3.85	219	649.4	3.94	232	656.3	4.02	
115	63.6	174	623.4	3.31	185	628.2	3.37	197	636.7	3.44	209	642.5	3.51	221	649.2	3.59	234	656.3	3.67	247	663.2	3.74	
125	68.1	188	629.2	3.09	199	635.5	3.16	211	642.2	3.22	224	648.6	3.28	236	655.7	3.36	249	662.6	3.42	263	670.2	3.50	
140	74.5	204	639.9	2.83	216	646.3	2.89	227	653.4	2.95	239	660.5	3.01	252	667.5	3.08	264	674.6	3.13	289	679.5	3.20	
155	80.4	219	644.7	2.62	231	651.8	2.67	242	658.3	2.72	256	666.2	2.78	269	672.2	2.84	279	679.8	2.89	296	687.8	2.95	
170	85.9	233	652.1	2.42	245	658.5	2.48	257	665.5	2.53	271	672.8	2.58	284	680.4	2.63	297	687.8	2.68	312	695.6	2.73	
190	92.7	251	660.6	2.22	264	667.9	2.26	276	674.8	2.30	289	682.3	2.36	303	689.8	2.41	317	697.7	2.46	332	705.7	2.51	
210	98.9	266	667.9	2.04	279	675.1	2.09	292	682.4	2.12	306	690.4	2.16	320	698.4	2.21	334	706.2	2.25	348	714.2	2.30	
230	104.7	281	675.4	1.90	295	683.1	1.94	308	690.9	1.98	321	698.2	2.02	336	706.2	2.07	350	714.1	2.11	365	722.2	2.15	
250	110.1	293	682.3	1.77	309	690.0	1.81	322	697.0	1.84	334	704.0	1.87	352	714.2	1.93	364	721.4	1.97	378	729.0	2.50	

Properties of Saturated Carbon Dioxide, CO_2

Based on Curves of the Institution of Mechanical Engineers of Great Britain and Work of Mollier

Temperature Deg. F.	Pressure, Lbs. per Sq.in.	Heat Content of Liquid, B.t.u. per Lb.	Heat of Vaporization, B.t.u. per Lb.	Heat Content of Vapor, B.t.u. per Lb.	Internal Latent Heat, B.t.u. per Lb.	External Work B.t.u. per Lb.	Entropy of Liquid.	Entropy of Vaporization.	Entropy of Vapor.	Specific Volume of Liquid, Cu.ft. per Lb.	Specific Volume of Vapor, Cu.ft. per Lb.	Specific Weight of Vapor, Lbs. per Cu.ft.	Temperature.
t	p'	i'	r	i''	ρ	ψ	s'	r/T	s''	v''	v''	m	t
−25	202	−22.2	124.7	102.5	108.1	16.6	−0.050	0.287	0.237	0.015	0.459	2.17	−25
−20	218	−20.5	123.2	102.7	107.1	16.1	−0.046	0.280	0.234	0.016	0.416	2.42	−20
−15	238	−18.8	121.6	102.8	105.5	16.1	−0.042	0.274	0.232	0.016	0.381	2.62	−15
−10	260	−17.1	120.0	102.9	104.0	16.0	−0.039	0.267	0.228	0.016	0.348	2.87	−10
−5	283	−15.3	118.3	103.0	102.4	15.9	−0.034	0.260	0.226	0.016	0.320	3.12	−5
0	308	−13.5	116.5	103.0	100.7	15.8	−0.030	0.254	0.224	0.016	0.293	3.41	0
5	334	−11.6	114.7	103.1	99.1	15.6	−0.027	0.248	0.221	0.016	0.268	3.73	5
10	362	−9.7	112.7	103.0	97.4	15.3	−0.022	0.240	0.218	0.017	0.245	4.08	10
15	391	−7.7	110.6	102.9	95.6	15.0	−0.017	0.232	0.215	0.017	0.224	4.46	15
20	422	−5.6	108.3	102.7	93.6	14.7	−0.013	0.225	0.212	0.017	0.205	4.88	20
25	454	−3.5	105.8	102.3	91.4	14.4	−0.008	0.217	0.209	0.017	0.188	5.32	25
30	489	−1.1	102.9	101.8	88.9	14.0	−0.003	0.209	0.206	0.017	0.172	5.81	30
35	526	1.5	99.7	101.2	86.1	13.6	+0.002	0.200	0.202	0.018	0.157	6.37	35
40	565	4.3	96.1	100.4	82.9	13.2	0.007	0.191	0.198	0.018	0.144	6.94	40
45	606	7.4	92.0	99.4	79.2	12.8	0.012	0.181	0.193	0.018	0.132	7.58	45
50	650	10.8	87.4	98.2	75.2	12.2	0.017	0.172	0.189	0.019	0.120	8.33	50
55	696	14.3	82.6	96.9	71.0	11.6	0.023	0.161	0.184	0.019	0.109	9.17	55
60	744	17.9	77.4	95.3	66.5	10.9	0.030	0.149	0.179	0.020	0.099	10.10	60
65	795	21.9	71.4	93.3	61.3	10.1	0.037	0.137	0.174	0.020	0.089	11.24	65
70	848	26.4	64.7	91.1	55.4	9.3	0.045	0.123	0.168	0.021	0.080	12.50	70
75	904	31.5	56.8	88.3	48.6	8.2	0.055	0.106	0.161	0.022	0.071	14.08	75
80	962	37.2	47.5	84.7	40.7	6.8	0.065	0.089	0.154	0.024	0.062	16.13	80
85	1022	45.4	33.8	79.2	28.5	5.3	0.080	0.061	0.141	0.026	0.054	18.52	85
87	1048	52.0	22.8	74.8	19.7	3.1	0.091	0.040	0.131	0.028	0.044	22.73	87
88.4	1070	63.0	0.0	63.0	0.0	0.0	0.112	0.000	0.112	0.035	0.035	28.57	88.4

Properties of Superheated Carbon Dioxide, CO_2

Based on Curves of Institution of Mechanical Engineers.

Pressure, Lbs. per per Sq.in.	Saturation Temp. Deg. F.	Entropy. 1.18			0.19			0.20			0.21			0.22		
		Temp. Deg. F.	Heat Content.	Specific Volume.	Temp. Deg. F.	Heat Content.	Specific Volume.	Temp. Deg. F.	Heat Content.	Specific Volume.	Temp. Deg. F.	Heat Content.	Specific Volume.	Temp. Deg. F.	Heat Content.	Specific Volume.
200	−25															
220	−19															
240	−14															
260	−10															
280	−6															
300	−2															
320	2															
340	6	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	8.6	103.5	0.270
360	10	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	14.0	104.4	0.250
380	13	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	21.2	105.3	0.235
400	16.5	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	28.4	106.0	0.222
450	24.1	.....	.....	.....	.....	.....	.....	.....	.....	.....	26.6	102.8	0.189	44.6	107.6	0.203
500	31.1	.....	.....	.....	.....	.....	.....	.....	.....	.....	41.0	104.4	0.178	60.8	109.6	0.195
550	38	.....	.....	.....	.....	.....	.....	39.2	100.8	0.147	55.4	105.8	0.164	73.4	111.2	0.181
600	44	.....	.....	.....	.....	.....	.....	51.8	102.1	0.139	68.0	107.3	0.154	86.0	112.7	0.169
650	50	.....	.....	.....	51.8	98.3	0.120	64.4	103.5	0.133	80.6	108.7	0.148	96.8	114.5	0.161
700	55	.....	.....	.....	62.6	99.5	0.114	77.0	104.8	0.127	91.4	110.3	0.138	107.6	115.9	0.150
750	61	62.6	95.4	0.100	73.4	100.6	0.108	87.8	105.8	0.118	100.4	111.6	0.129	116.6	117.2	0.140
800	66	71.6	96.3	0.092	82.4	101.9	0.104	95.0	107.1	0.113	109.4	112.9	0.123	122.0	118.4	0.130
850	70	80.6	97.2	0.089	91.4	102.6	0.099	104.0	108.0	0.107	118.4	114.1	0.116	132.8	120.1	0.125
900	75	89.6	98.1	0.081	100.4	103.9	0.095	113.0	109.4	0.104	127.4	115.2	0.113	141.8	121.1	0.120
1000	83	100.4	99.7	0.076	113.0	105.7	0.086	127.4	111.2	0.095	140.0	117.0	0.103	158.0	123.3	0.112
1070	88															
1100	.....	116.6	101.3	0.074	127.4	107.1	0.080	140.0	113.0	0.088	154.4	119.2	0.095	172.4	125.5	0.104
1200	.....	129.2	102.6	0.070	141.8	108.5	0.077	154.4	114.8	0.084	170.6	120.8	0.091	186.8	127.4	0.098

PROPERTIES OF SUPERHEATED CARBON DIOXIDE, CO_2—*Continued*

Pressure, Lbs. per Sq.in.	Saturation Temp. Deg. F.	Entropy. 0.23 Temp. Deg. F.	0.23 Heat Content.	0.23 Specific Volume.	0.24 Temp. Deg. F.	0.24 Heat Content.	0.24 Specific Volume.	0.25 Temp. Deg. F.	0.25 Heat Content.	0.25 Specific Volume.
200	−25	.....	.....	.....	−16.6	103.3	0.460	+7.2	107.5	0.500
220	−19	.....	.....	.....	−9.4	104.9	0.420	13.1	109.6	0.460
240	−14	.....	.....	.....	1.4	106.6	0.385	23.0	111.2	0.420
260	−10	−10.0	103.7	0.348	15.8	108.0	0.365	35.6	112.9	0.395
280	−6	+1.4	104.9	0.318	23.0	109.6	0.344	46.4	114.5	0.370
300	−2	10.4	106.2	0.301	32.0	110.9	0.330	55.4	115.9	0.355
320	2	19.4	107.3	0.288	42.8	112.1	0.316	64.4	117.2	0.340
340	6	28.4	108.4	0.275	51.8	113.4	0.305	73.4	118.6	0.328
360	10	35.6	109.4	0.263	59.0	114.7	0.288	81.5	119.9	0.318
380	13	42.8	110.3	0.250	64.4	115.4	0.278	87.8	120.8	0.300
400	16	50.0	111.2	0.246	71.6	116.3	0.263	93.2	121.7	0.278
450	24	66.2	113.2	0.226	87.8	118.4	0.248	104.0	124.2	0.261
500	31	82.4	115.2	0.210	98.6	120.6	0.223	116.6	126.7	0.238
550	38	93.2	117.0	0.198	111.2	122.8	0.210	125.6	129.1	0.220
600	44	104.0	118.8	0.182	118.4	124.6	0.194	138.2	131.0	0.207
650	50	114.8	120.4	0.173	129.2	126.4	0.183	149.0	132.8	0.196
700	55	123.8	122.0	0.160	140.0	128.2	0.170	158.8	134.5	0.183
750	61	131.0	123.5	0.150	150.8	129.6	0.161	170.6	135.5	0.175
800	66	141.8	124.6	0.142	159.8	130.9	0.153	181.4	137.0	0.165
850	70	150.8	126.0	0.135	170.6	132.3	0.145	190.4	138.6	0.155
900	75	159.8	127.6	0.130	188.6	133.7	0.138	201.2	140.2	0.152
1000	83	177.8	129.8	0.120	199.4	136.3	0.130	219.2	142.7	0.138
1070	88									
1100	.....	194.0	131.9	0.113	205.6	138.6	0.117	230.0	144.9	0.127
1200	.....	208.4	134.3	0.106	228.2	140.8	0.115			

Properties of Sulphur Dioxide, SO_2

Based on Curves of the Institution of Mechanical Engineers of Great Britain and Work of Lange

Temperature Degrees F.	Pressure, Lbs. per Sq.in.	Heat Content of Liquid, B.t.u. per Lb.	Heat of Vaporization, B.t.u. per Lb.	Heat Content of Dry Vapor, B.t.u. per Lb.	Internal Latent Heat, B.t.u. per Lb.	External Work B.t.u. per Lb.	Entropy of Liquid.	Entropy of Vaporization.	Entropy of Vapor.	Specific Volume of Liquid, Cu.ft. per Lb.	Specific Volume of Vapor, Cu.ft. per Lb.	Specific Weight of Vapor, Lbs. per Cu.ft.	Temperature, Degrees F.
t	p	i'	r	i''	ρ	ψ	s'	r/t	s''	v'	v''	m	t
−25	5.1	−17.4	165.5	148.1	135.0	13.1	−0.038	0.380	0.342	0.01	13.89	0.072	−25
−20	5.9	−15.9	164.9	149.0	135.9	13.1	−0.035	0.375	0.340	0.01	12.02	0.083	−20
−15	6.8	−14.4	164.2	149.8	136.6	13.2	−0.031	0.369	0.338	0.01	10.42	0.096	−15
−10	7.9	−12.9	163.6	150.7	137.4	13.3	−0.028	0.364	0.336	0.01	9.12	0.110	−10
− 5	9.1	−11.4	162.8	151.4	138.1	13.3	−0.025	0.358	0.333	0.01	8.05	0.124	−5
0	10.4	−9.9	162.0	152.1	138.7	13.4	−0.021	0.352	0.331	0.01	7.12	0.140	0
5	11.8	−8.3	161.3	153.0	139.5	13.5	−0.018	0.347	0.329	0.01	6.27	0.159	5
10	13.3	−6.8	160.5	153.7	140.1	13.6	−0.015	0.342	0.327	0.01	5.56	0.180	10
15	15.0	−5.3	159.7	154.4	140.7	13.7	−0.011	0.336	0.325	0.01	4.95	0.202	15
20	17.0	−3.8	159.1	155.3	141.5	13.8	−0.008	0.332	0.324	0.01	4.42	0.225	20
25	19.2	−2.2	158.3	156.1	142.1	14.0	−0.005	0.326	0.321	0.01	3.96	0.253	25
30	21.5	−0.6	157.4	156.8	142.5	14.1	−0.001	0.321	0.320	0.01	3.56	0.281	30
35	24.0	1.0	156.3	157.3	143.1	14.2	0.002	0.316	0.318	0.01	3.20	0.312	35
40	26.7	2.6	155.4	158.0	143.7	14.3	0.005	0.311	0.316	0.01	2.88	0.347	40
45	29.8	4.3	154.3	158.6	144.2	14.4	0.008	0.306	0.314	0.01	2.61	0.383	45
50	33.0	6.0	153.3	159.3	144.8	14.5	0.012	0.301	0.313	0.01	2.36	0.424	50
55	36.5	7.7	152.1	159.8	145.3	14.5	0.015	0.296	0.311	0.01	2.15	0.465	55
60	40.4	9.3	151.1	160.4	145.8	14.6	0.018	0.291	0.309	0.01	1.96	0.510	60
65	44.7	11.0	149.9	160.9	146.2	14.7	0.021	0.286	0.307	0.01	1.78	0.562	65
70	49.2	12.7	148.8	161.5	146.8	14.7	0.025	0.281	0.306	0.01	1.62	0.617	70
75	54.0	14.4	147.6	162.0	147.2	14.8	0.028	0.276	0.304	0.01	1.48	0.676	75
80	59.3	16.1	146.4	162.5	147.7	14.8	0.031	0.271	0.302	0.01	1.37	0.730	80
85	64.9	17.8	145.1	162.9	148.0	14.9	0.034	0.266	0.300	0.01	1.25	0.800	85
90	70.9	19.5	143.9	163.4	148.5	14.9	0.037	0.262	0.299	0.01	1.15	0.870	90
95	77.5	21.3	142.5	163.8	148.8	15.0	0.041	0.257	0.298	0.01	1.05	0.952	95
100	84.4	23.1	140.9	164.0	149.0	15.0	0.044	0.252	0.296	0.01	0.96	1.042	100
105	91.8	24.8	139.4	164.2	149.3	14.9	0.047	0.247	0.294	0.01	0.88	1.136	105
110	99.4	26.6	137.7	164.4	149.5	14.9	0.050	0.242	0.292	0.01	0.82	1.220	110
115	107.2	28.4	136.3	164.7	149.8	14.9	0.053	0.237	0.290	0.01	0.77	1.299	115

PROPERTIES OF SUPERHEATED SULPHUR DIOXIDE, SO_2

Pressure Lbs. per Sq.in.	Saturation Temperature.	Entropy. 0.30			0.31			0.32			0.33			0.34			0.35			0.36		
		Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.
6	−19.5	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	−4	154.1	12.48	12	158.4	12.75
7	−13.9	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	−9	150.8	10.86	3	155.5	11.00	18	160.2	11.14
8	−9.5	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	−4	152.5	9.37	10	157.1	9.57	25	162.0	9.76
9	−5.5	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	+1	153.9	8.36	16	158.4	8.54	28	163.4	8.73
10	−1.5	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	7	155.2	7.54	21	160.0	7.69	36	164.9	7.32
11	2.5	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	13	156.6	6.96	27	161.6	7.08	41	166.1	7.19
12	5.7	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	18	157.9	6.41	32	162.7	6.53	46	167.6	6.65
13	9.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	10	154.3	5.77	23	158.9	5.91	37	164.0	6.05	52	169.0	6.18
14	12.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	14	155.3	5.34	27	160.2	5.48	43	165.2	5.62	57	170.5	5.75
15	15.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	18	156.2	5.05	32	161.3	5.15	46	166.5	5.25	61	171.5	5.35
16	17.5	.....	.....	.....	.....	.....	.....	.....	.....	.....	23	157.3	4.75	36	162.2	4.84	50	167.2	4.93	65	172.4	5.03
17	20.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	27	157.2	4.49	41	163.3	4.58	55	168.1	4.67	71	173.3	4.76
18	22.3	.....	.....	.....	.....	.....	.....	.....	.....	.....	30	158.9	4.27	45	164.0	4.35	59	169.0	4.44	75	174.2	4.52
19	24.5	.....	.....	.....	.....	.....	.....	.....	.....	.....	34	159.8	4.06	48	164.8	4.14	63	169.9	4.23	79	175.1	4.31
20	26.8	.....	.....	.....	.....	.....	.....	.....	.....	.....	37	160.6	3.87	52	165.6	3.95	66	170.8	4.02	84	176.0	4.10
22	31.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	43	161.8	3.55	57	166.9	3.63	73	172.1	3.71	90	177.5	3.77
24	35.0	.....	.....	.....	.....	.....	.....	36	158.0	3.20	48	163.1	3.27	63	168.1	3.35	79	173.3	3.43	95	178.9	3.49
26	38.6	.....	.....	.....	.....	.....	.....	41	159.1	2.96	54	164.3	3.03	70	169.4	3.10	84	175.0	3.17	100	180.4	3.23
28	42.2	.....	.....	.....	.....	.....	.....	46	160.4	2.83	59	165.6	2.88	75	170.8	2.93	91	176.2	2.97	107	182.0	3.02
30	45.3	.....	.....	.....	.....	.....	.....	52	161.6	2.60	64	166.9	2.65	82	172.3	2.71	99	177.8	2.77	113	183.6	2.82

PROPERTIES OF SUPERHEATED SULPHUR DIOXIDE, SO_2—*Continued*

Pressure, Lbs. per Sq. in.	Saturation Temperature.	Entropy.																					
		0.30			0.31			0.32			0.33			0.34			0.35			0.36			
		Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	Temperature.	Heat Content.	Specific Volume.	
32	48.5	.....	.....	.....	.....	.....	.....	57	162.5	2.44	72	167.8	2.49	86	173.3	2.54	104	179.2	2.60	118	185.0	2.66	
34	51.4	.....	.....	.....	.....	.....	.....	61	163.5	2.31	77	168.8	2.35	91	174.6	2.41	108	180.2	2.47	126	186.1	2.51	
36	54.3	.....	.....	.....	.....	.....	.....	66	164.5	2.20	81	169.9	2.24	95	175.5	2.28	111	181.3	2.33	129	187.2	2.37	
38	57.0	.....	.....	.....	.....	.....	.....	60	165.2	2.09	84	170.8	2.13	100	176.4	2.17	117	182.2	2.22	135	188.1	2.26	
40	59.5	.....	.....	.....	61	160.9	1.97	75	166.1	2.00	88	171.5	2.04	104	176.3	2.08	120	183.1	2.12	138	189.0	2.16	
42	61.8	.....	.....	.....	64	161.6	1.88	79	166.9	1.92	91	172.3	1.95	108	178.0	1.99	126	184.0	2.03	144	189.9	2.07	
44	64.2	.....	.....	.....	68	162.4	1.81	82	167.6	1.84	95	173.2	1.87	112	178.9	1.91	129	184.9	1.94	147	190.8	1.98	
47	67.5	.....	.....	.....	72	163.1	1.70	86	168.5	1.74	100	174.1	1.76	117	180.0	1.80	134	185.9	1.82	151	192.1	1.86	
50	69.9	.....	.....	.....	77	164.0	1.61	91	169.4	1.64	108	175.1	1.66	122	181.3	1.69	140	187.2	1.73	156	193.3	1.76	
55	76.0	.....	.....	.....	84	165.6	1.47	97	171.0	1.50	115	176.6	1.53	131	182.9	1.56	147	189.0	1.59	163	195.1	1.61	
60	80.6	.....	.....	.....	91	167.0	1.36	106	172.6	1.38	122	178.2	1.40	138	184.5	1.43	154	190.8	1.46	172	197.1	1.48	
65	85.1	.....	.....	.....	99	168.1	1.26	113	174.0	1.28	129	179.8	1.30	145	186.1	1.33	163	192.2	1.36				
70	89.3	91	163.8	1.15	106	169.4	1.18	120	175.1	1.19	136	181.3	1.22	153	187.4	1.24	170	198.9	1.26				
75	93.0	97	164.7	1.08	111	170.6	1.10	126	176.4	1.12	144	182.3	1.14	160	188.8	1.16	176	195.3	1.18				
80	97.0	103	165.6	1.02	117	171.5	1.05	133	177.3	1.07	149	183.6	1.08	167	190.1	1.10	185	196.6	1.12				
85	100.4	108	166.5	0.97	122	172.4	0.99	136	178.2	1.01	154	184.7	1.02	172	191.3	1.04	190	198.0	1.06				
90	103.8	112	167.2	0.91	127	173.3	0.94	144	179.5	0.95	162	185.9	0.97	180	192.4	0.99							
95	107.1	116	167.9	0.87	131	174.1	0.90	149	180.3	0.92	167	186.8	0.93	185	193.3	0.95							
100	109.9	120	168.7	0.84	136	174.8	0.86	153	181.3	0.87	172	187.7	0.89	192	194.4	0.92							

TABLE OF CHLORIDE OF CALCIUM ($CaCl_2$) SOLUTION

Specific Gravity at 60 F.	Degrees Beaumé at 60° F.	Degrees Salometer at 60° F.	Lbs. ($CaCl_2$) per gallon Solution.	Lbs. ($CaCl_2$) per Cu. ft. Solution.	Percentage ($CaCl_2$) by Weight.	Freezing-point F.	Specific Heat at 32 F.	Weight per Gallon at 60° F.
1.021	3	12	$\frac{1}{2}$	$3\frac{3}{4}$	3	+29	0.965	8.54
1.043	6	27	1	$7\frac{1}{2}$	5	+27	0.920	8.70
1.066	9	36	$1\frac{1}{4}$	$8\frac{1}{2}$	7	+25	0.883	8.88
1.074	10	40	$1\frac{1}{2}$	$11\frac{1}{4}$	9	+23	0.868	8.96
1.082	11	44	$1\frac{3}{4}$	13	10	+21	0.857	9.05
1.099	13	52	2	15	12	+18	0.830	9.19
1.115	15	62	$2\frac{1}{4}$	17	14	+14	0.808	9.29
1.160	20	80	$2\frac{1}{2}$	19	18	+4	0.753	9.65
1.179	22	88	3	$22\frac{1}{2}$	20	−1.5	0.732	9.83
1.198	24	95	$3\frac{1}{2}$	26	22	−8	0.714	10.00
1.219	26	104	4	30	24	−17	0.695	10.16
1.239	28	112	$4\frac{1}{2}$	34	26	−27	0.678	10.32
1.261	30	120	5	$37\frac{1}{2}$	28	−39	0.661	10.50
1.283	32	128	$5\frac{1}{2}$	$41\frac{1}{2}$	30	−54	0.643	10.72

If more chloride is used the freezing-point is raised. Use about 1 ton of $CaCl_2$ for each ton of ice-making capacity.

TABLE OF SODIUM CHLORIDE (SALT) SOLUTION

Specific Gravity at 39° F.	Degrees Beaumé at 60° F.	Degrees of Salometer at 60 °F.	Pounds of Salt per Gallon of Solution.	Pounds of Salt per Cu.ft.	Percentage of Salt by Weight.	Freezing-point Fahrenheit.	Specific Heat.	Weight per Gallon at 39° F.
1.007	1	4	0.084	.628	1	31.8	0.992	8.40
1.015	2	8	0.169	1.264	2	29.3	0.984	8.46
1.023	3	12	0.256	1.914	3	27.8	0.976	8.53
1.030	4	16	0.344	2.573	4	26.6	0.968	8.59
1.037	5	20	0.433	3.238	5	25.2	0.960	8.65
1.045	6	24	0.523	3.912	6	23.9	0.946	8.72
1.053	7	28	0.617	4.615	7	22.5	0.932	8.78
1.061	8	32	0.708	5.295	8	21.2	0.919	8.85
1.068	9	36	0.802	5.998	9	19.9	0.905	8.91
1.076	10	40	0.897	6.709	10	18.7	0.892	8.97
1.091	12	48	1.092	8.168	12	16.0	0.874	9.10
1.115	15	60	1.389	10.389	15	12.2	0.855	9.26
1.155	20	80	1.928	14.421	20	6.1	0.829	9.64
1.187	24	96	2.376	17.772	24	1.2	0.795	9.90
1.196	25	100	2.488	18.610	25	0.5	0.783	9.97
1.204	26	104	2.610	19.522	26	1.1	0.771	10.04

Correction for temperature of aqua ammonia to reduce Beaumé readings to 60° readings subtract $\frac{1}{4}$° Beaumé for the following number of degrees F:

From 18 to 20° B. for each 8° F. above 60° F.
From 20 to 22° B. for each 7° F. above 60° F.
From 22 to $23\frac{1}{2}$° B. for each 6° F. above 60° F.
From $23\frac{1}{2}$ to $25\frac{1}{4}$° B. for each 5° F. above 60° F.
Above this for each 4° F. above 60° F.

COMPARISON OF THERMOMETERS

Cent.	Fahr.	Cent.	Fahr.	Cent.	Fahr.
−40	−40.0	8	46.4	56	132.8
−38	−36.4	10	50.0	58	136.4
−36	−32.8	12	53.6	60	140.0
−34	−29.2	14	57.2	62	143.6
−32	−25.6	16	60.8	64	147.2
−30	−22.0	18	64.4	66	150.8
−28	−18.4	20	68.0	68	154.4
−26	−14.8	22	71.6	70	158.0
−24	−11.2	24	75.2	72	161.6
−22	−7.6	26	78.8	74	165.2
−20	−4.0	28	82.4	76	168.8
−18	−0.4	30	86.0	78	172.4
−16	+3.2	32	89.6	80	176.0
−14	6.8	34	93.2	82	179.6
−12	10.4	36	96.8	84	183.2
−10	14.0	38	100.4	86	186.8
−8	17.6	40	104.0	88	190.4
−6	21.2	42	107.6	90	194.0
−4	24.8	44	111.2	92	197.6
−2	28.4	46	114.8	94	201.2
0	32.0	48	118.4	96	204.8
2	35.6	50	122.0	98	208.4
4	39.2	52	125.6	100	212.0
6	42.8	54	129.2		

TESTING REFRIGERATING APPARATUS

Tests of refrigerating apparatus are difficult to perform because the changes of temperature in various parts of the apparatus are very slight, because the weight and quality of the refrigerating medium is difficult to determine and because the

errors at start and finish of the test make it necessary to carry the test over a considerable time.

Tests are necessary to determine the effects or values of new devices and alterations and particularly to determine whether or not the guaranteed amount of refrigeration or the guaranteed refrigerating effect has been obtained.

To find the yield of the apparatus, the refrigerating effect may be measured from the ammonia or from the brine or in an ice plant the amount of ice produced may be found. If the guarantee is the production of a certain amount of ice or the cooling of certain rooms to a definite temperature with a given amount of power and cooling water this test is simple except for the length of test, which should never be less than twenty-four hours and would be much better if continued for one or two weeks. When, however, the refrigerating effect is to be found the test is difficult because of the quantities to be measured.

The refrigeration produced from the ammonia is given by

$Q_r = M(i_1 - i_3)$;
M = weight of ammonia in given time;
i_1 = heat content in suction main leaving expansion coil;
i_3 = heat content at entrance to expansion valve.

To determine this, the various factors on the right-hand side of the equation must be found. The weight of ammonia, M, may be found by collecting the ammonia in a receiver resting on a platform scale. This is connected to the piping system by a long piece of pipe so that there will be only a slight effect from the rigidity of the pipes. If the pipes are 10 ft. long the weight necessary to deflect the pipe an amount equal to the movement of the scale platform will be so small that little error results. By using two receivers, one may be filling while the other is being emptied. The ammonia may be collected in two tanks and the volumes measured, this being changed to weight by calibration. Care must be taken to have no pockets in the piping in which the ammonia may collect. In fact the uncertainty of the amount of ammonia which may lodge in pockets makes this method a difficult one and for that

reason in some tests, such as those at the Eastman Kodak Co., the refrigeration is measured from the brine side.

The quantities i_1 and i_3 are difficult to determine if by chance there are liquid and vapor present together. To prevent this, the liquid going to the expansion valve is after-cooled so that it is below the temperature of vaporization corresponding to the pressure and must be all liquid (hence $i_3 = q_3'$) and the vapor entering the suction valve is slightly superheated so that the quality may be determined by a thermometer

$$(i_1 = q' + r + c_p(t_{\text{sup}} - t_{\text{sat}})).$$

In this way it is possible to find the value of i_1 and i_3. The compression is dry compression. If it is desired to have wet compression it would be possible to have slightly superheated ammonia in the suction pipe and add an amount of liquid ammonia from a calibrated tank.

The thermometers placed in the thermometer wells are subject to errors, due to the warming of the stem if any mercury projects above the well, and if none projects above there is difficulty in reading the thermometer, as the stem often freezes fast to the well on the suction side. To correct for stem error it is well to determine the temperature of the stem t_s by a small thermometer tied to the stem and if t_t is the reading of the thermometer, and t_w is the reading of the point of the thermometer opposite the edge of the well so that the number of degrees exposed, above the well is $t_t - t_w$, the correction to be added to the reading is

$$\Delta t = 0.000088(t_t - t_s)(t_t - t_w).$$

This assumes that the length $(t_t - t_w)$ is heated $(t_t - t_s)$ degrees above the temperature it should have been, and 0.000088 is the difference between the coefficient of expansion of glass and mercury. Constant immersion thermometers which have been calibrated to read correctly in rooms of a certain temperature when immersed to a mark on the stem may be used with no correction.

To save corrections and troubles in observation, thermo couples or resistance thermometers may be used.

Calibrated gauge readings as well as temperatures are necessary on suction and discharge to determine the quantities i_1 and i_3. The suction pressure is often measured by using a mercury U-tube so that small pressure differences may be read.

On account of the errors in the method above, the refrigeration is sometimes determined from the brine, which is cooled by the ammonia.

$Q_r = M_b c(t_0 - t_1)$;
M_b = weight of brine;
c = specific heat of brine;
t_0 = temperature of brine at outlet from cooler;
t_i = temperature of brine at inlet to cooler.

In this case the weight of brine M_b is measured by weighing in large tanks; by the use of a meter which is calibrated at intervals during test; by the use of a Venturi meter or weir. The calibrations of these pieces of apparatus are absolutely necessary.

The specific heat c must be determined by a formula from the specific gravity of the brine and the temperature or by tables given earlier or, what is better, by an experimental determination made in a Dewar flask by finding the watt hours used in a coil and required to warm a certain amount of brine between the temperatures used during the test. Corrections can be made for radiation by cooling curves and the water equivalent may be found by method of mixtures or by heating distilled water. In finding c by the formula or table the mean temperature is used and the specific gravity is determined by a hydrometer of some form.

The temperatures t_0 and t_i are subject to the same corrections mentioned before and because the difference between them is so small the thermometer should be graduated to tenths of a degree, or smaller, divisions.

The use of Beckman thermometers would prove of value here.

The power of the engine required to drive the compressor and the power of the compressor are determined by indicator diagrams. The small clearance on the compressor makes it necessary to use close and small connections for the indicators as the increase of clearance when the indicator is opened will change the form of card. It would be well to have a compensating volume to cut out when the indicator is connected or the indicator might be placed so that the piston would move vertically downward and the passage from the cylinder could be filled with oil so that this volume is not filled with ammonia. The usual formula for horse-power is used.

It is necessary to test the springs and have the reducing motion correct.

The amount of cooling water is weighed, metered, passed through an orifice or over a weir and its temperature is determined by thermometers. The heat is given by

$$Q_c = M_w(i'_0 - i'_i).$$

In all cases the machines should be brought to an operating condition before starting the test and a running start should be made. With brine to determine refrigeration ten hours after steady conditions are obtained may be sufficient, although with ammonia a longer test must be used. In testing absorption machines calibrated meters are used to determine the flow of liquor and the strength of the liquor is determined by drawing off samples and using a hydrometer to give the specific gravity. Thermometers and gauges give the conditions at the various points. Meters are used to measure the cooling water and thermometers give the temperatures from which the heat may be determined. The drip from the separator may be determined by passing it into one of two cylinders and measuring its volume on the way to the analyzer.

The test of ice plants should extend over a number of days five or seven, and in this test the ice is pulled at regular intervals during the twenty-four hours. The test is started after the plant has been run at least seventy-two hours to get steady conditions.

General observations should be made at fifteen-minute intervals. These include the following: Temperatures: outside atmosphere, engine room, refrigerated rooms, condensing water at inlet and outlet, brine at inlet to cooler and at outlet, ammonia at entrance to expansion valve and at entrance to suction main, at suction valve and at discharge main on compressor, at inlet and outlet to jacket; pressures on suction and discharge main, and in expansion coil, barometer; volume shown by meter on brine line, condensing water line and jacket water line, indicator cards, revolutions of compressor, weight of water going to ice tanks with temperature, weight of water left unfrozen, weight of coal, weight of boiler feed, feed temperature, calorimeter readings, flue gas temperature.

The computation for such a test will be given in the next chapter.

For absorption machines the readings are somewhat similar and are used in the same manner.

A form of test has been discussed by the A. S. R. E. in its proceedings.

The following data are obtained from a series of tests:

RESULTS OF TEST ON DOUBLE-ACTING COMPRESSOR, MADE BY THE DE LA VERGNE CO. AT THE EASTMAN KODAK CO., DATE FEB. 5, 1908

Temperature:	Discharge ammonia R.H	149.44° F.
	L.H	143.36° F.
	Suction at compressor, before liquid injection	17.80° F.
	At brine cooler	19.40° F.
	Before expansion valve	58.91° F.
	Brine at inlet to cooler	25.11° F.
	Brine at outlet from cooler	14.81° F.
	Engine room	65.85° F.
	Ammonia receiver room	55.58° F.
	Outside atmosphere	14.93° F.
Revolutions in 15-minute compressor		512.1
Revolutions in 15-minute brine pump		419

Specific heat of brine	0.678
Weight of brine per revolution	41.15 lbs.
Specific heat of liquid ammonia	1.1
Pounds of liquid ammonia in 15 min	236.6
Pressures: Suction at cooler	20.45
at compressor	20.03
at condenser	185.06
Steam at engine	84.11
Barometer	15.01
M.E.P. Head end	38.95
Crank end	39.67
I.H.P	55.83
Tons of refrigeration by brine	36.91
Equivalent tons with 20 lb. suction	36.88
I.H.P. per ton	1.514

Size of compressor . . . $11\frac{1}{4} \times 22 - (2\frac{1}{2}$ piston rod) double acting
Size of engine . . . . . 22×22 ($3\frac{5}{8}$ piston rod)

RESULTS OF TEST ON SINGLE-ACTING COMPRESSOR MADE BY THE YORK MFG. CO. AT THE EASTMAN KODAK CO., DATE MAR. 9, 1908.

Temperatures: Discharge ammonia R.H	248.3° F.
L.H	243.3° F.
Suction at compressor R.H	14.34° F.
L.H	15.20° F.
At cooler	9.29° F.
Before expansion valve	77.91° F.
Brine at inlet to cooler	22.73° F.
Brine at outlet from cooler	13.02° F.
Engine room	64.80° F.
Ammonia receiver room	73.46° F.
Outside atmosphere	24.79° F.
R.H. water jacket	180.7° F.
L.H. water jacket	168.45° F.
Revolutions in 15-minute compressor	514.7
Brine pump	426.85

Specific heat of brine		0.678
Weight of brine per revolution		41.15 lb.
Pounds of liquid ammonia in 15 minutes		233.9
Pressures: Suction at cooler		20.46
	at compressor	20.04
	at condenser	187.27
	Steam at engine	81.96
	Barometer	14.95
M.E.P.: Head end		36.56
Crank end		37.09
I.H.P.		52.57
Tons of refrigeration by brine		37.01
Equivalent tons with 20-lb. suction		36.97
I.H.P. per ton		1.42
Size of compressor		15×22 (single acting)
Size of engine		22×22 ($3\frac{5}{8}$ piston rod)

TEST OF TWO DE LA VERGNE STANDARD HORIZONTAL REFRIGERATING MACHINES, DATE OCT. 27, 1910

Temperatures: Ammonia discharge		245.80° F.
Ammonia suction at brine coolers		5.0° F.
Ammonia before expansion valves		8.14° F.
Brine at inlet		6.20° F.
Brine at outlet		17.6° F.
Revolutions of compressors		43.5
Pressures: Suction		16.23
At condensers		165.50
Total horse-power		517.88
Tons of refrigeration		373.23
I.H.P. per ton		1.398
Size of compressors double acting		$18\frac{1}{4}''\times33''$
Size of engine		22 & 44×33
Volumetric eff.: Apparent		95.38
True		82.15
Rated capacity, two, 275 tons, 550 tons		

Test of Kroeschell Bros. Ice Machine Co.'s CO_2 Compressor, Date August 9, 1907

Compressor, double acting, horizontal.

Bore	138 $M/M = 5\frac{7}{16}''$
Stroke	508 $M/M = 20''$
Piston rod	58 $M/M = 2\frac{5}{16}''$
Speed	65 R.P.M.
Compressor gas displacement	54,914.6 cu.in. per min.
Condenser pressure	65 atm.
Evaporating pressure	23 atm.
Evaporating temperature	5° F.
Water temperature condenser inlet	53° F.
Water temperature condenser outlet	81° F.
Temperature of brine cooler inlet	25.9° F.
Temperature of brine cooler outlet	17° F.
Quantity of brine pumped per hour	1000 cu.ft.
Strength of brine	26° Beaumé
Estimated loss in brine tanks and cooler	10%
Amount of refrigeration	50.76 tons
Indicated power at engine	67.25 H.P.
Indicated power at compressor	51.3 H.P.
Compressor gas displacement per ton per per min	1082 cu.in.
Horse-power ÷ cooling effect = 1.010 H.P. per ton of refrigeration.	

Test of Vogt Absorption Plant

Aqua ammonia pump	$5\frac{1}{2} \times 12$
Average speed pump	22 R.P.M.
Temperature: Brine inlet	15° F.
Brine outlet	13° F.
Ammonia at condenser	105° F.
Liquid from condenser	78° F.
Strong aqua to rectifier	89° F.
Strong aqua from rectifier	120° F.
Strong aqua from exchanges	189° F.

Weak aqua from cooler	89° F.
Cooling water	65° F.
Cooling water from condenser	72° F.
Cooling water from absorber	88° F.
Cooling water from weak aqua cooler	117°
Strong aqua at 60° F	$26\frac{1}{2}$° Beaumé
Weak aqua at 60° F	$23\frac{1}{2}$° Beaumé
Total cooling water per min.	252 gals.
Ice per ton of coal	10.3 tons

TEST OF WESTINGHOUSE-LEBLANC REFRIGERATING MACHINES, DATE AUG. 4TH AND 5TH, 1916

Barometer		29.17″
Temperature atmosphere		86° F.
Live steam pressure		200 lbs. per sq.in.
Vacuum:	1st ejector	29
	2d ejector	28.95
	Condenser	27.80
Temperature:	Condensing water inlet	82.5° F.
	Condensing water outlet	92° F.
	Brine inlet	18.40° F.
	Outlet	15.00° F.
Weight of brine per hour		19826
Specific heat brine		.833
B.t.u. of refrigeration per hour		56,202
Tons per twenty-four hours		4.60
Loss by radiation		.37
Total loss		5.05
Steam and vapor condensed per hour		1125
Vapor per hour		61
Steam per hour		1064

Tons of refrigeration per ton of coal at 8 lbs. of steam per pound of coal:

$$\frac{\frac{5.05}{3}}{\frac{1064\times 24}{8\times 2000}} = \frac{5.05}{1.596} = 3.17$$

TEST OF WESTINGHOUSE-LEBLANC APPARATUS FOR COOLING WATER. MARCH 24, 1915

Barometer	29.18
Vacuum in 1st ejector	28.91
Vacuum in condenser	27.40
Steam pressure (gauge)	125
1st ejector	112.5
2d ejector	110.5
Quality	98.7%
Temperatures:	
Water to be cooled (brine ordinarily)	
at inlet	40.5° F.
at outlet	34.9° F.
Steam, line	357.3° F.
Atmosphere	69.1° F.
Condenser, top	155.5° F.
Condensate	89.7° F.
Circulating water inlet	79.9° F.
Outlet	89.7° F.
Weights per hour:	
Water to be cooled	101,320
Circulating water	450,000
Condensed steam	2513
Heads pumped against	67.2 ft.
Power brine pump	8.83 H.P.
Condenser	35.7 H.P.
Refrigeration	567,392 B.t.u. per hr.

TESTS OF HART COOLING TOWER

Temperature.							Gallons per Min.	Rel. Humidity.
Water.			Air.		Above Wet Bulb.	Below Atmosphere.		
Entering.	Leaving.	Reduction.	Dry Bulb.	Wet Bulb.				
78	74	4	88	72	2	14	600	41
77	71	6	88	71	0	17	600	39
81	76	5	88	74	2	12	600	46
79	76	3	84	74	2	8	600	57
82	75	7	81	73	2	6	600	64
75	69	6	71	67	2	2	600	78
108	76	32	87	73	3	11	1800	46
111	73	36	76	66	7	3	1800	55
108	76	32	81	72	4	5	1800	60
109	78	31	79	73	5	1	1800	71
108	74	34	74	69	5	0	1800	74
83	68	15	77	62	6	9	3000	41

CHAPTER X

PROBLEMS

THIS chapter is devoted to problems illustrating the application of the text. They are typical problems, and the student is urged to consider them as illustrating principles so that other problems of similar nature may be solved in the same manner. Certain problems are solved for a given set of conditions and if these conditions change the results will, of course, differ from those obtained. The general problems of design are repetitions of certain fundamental problems and it has been the aim of the author to include these fundamentals in this set. The data to be used in actual problems must be obtained for the particular locality.

Problem 1. Find the **best thickness of cork insulation** for an 8-in. wall on a room which is to be held at 20° F.

Cork thicknesses possible: 2 in., 3 in., two 2 in., 2 and 3 in., two 3 in., etc.

Cost of 1 sq.ft. of cork installed (page 346):

2 in.	25 cts.
3 in.	30 cts.
4 in.	40 cts.
5 in.	50 cts.
6 in.	60 cts.

Value of 1 cu.ft. of storage space, per mo. . . . . 5 cts. (p. 215)
Cost of 1 ton of refrigeration . . . . . 40 cts. (p. 376).
Average outside temperature per year . . . . . 48° F. (Fig. 166).

Fixed charges (p. 379).

Interest	8%
Taxes	1%
Insurance	1%
Depreciation	3%
Repairs	2%
Total	15%

Coefficient for 8-in. walls with plaster.....$K=0.37$ (p. 211)
C for cork..... 0.022
C for plaster..... 0.46

For completed walls the effect of additional layers of materials may be computed in the following way:

$$K'=\frac{1}{\frac{1}{K}+\frac{l}{C}+\frac{l'}{C'}}.$$

K' = new constant;

K = former constant for wall;

l = thicknesses of new layers;

C = coefficients of new layers.

(*a*) K for 8″ brick, 2″ cork, 1″ plaster.

$$K'=\frac{1}{\frac{1}{0.37}+\frac{2}{12\times 0.022}+\frac{1}{12\times 0.46}}=0.095.$$

(*b*) K for 8″ brick, 3″ cork, 1″ plaster.

$$K'=\frac{1}{\frac{1}{0.37}+\frac{3}{12\times 0.022}+\frac{1}{12\times 0.46}}=0.070.$$

(*c*) K for 8″ brick, 4″ cork, 1″ plaster (2″, ½″, 2″, ½″).

$$K'=0.055.$$

(*d*) K for 8″ brick, 5″ cork, 1″ plaster (2″, ½″, 3″, ½″).

$$K'=0.046.$$

Heat Loss per Year in Tons of Refrigeration per square foot.

(*a*) $$\frac{365\times 24\times .095(48-20)}{28{,}680}=0.81 \text{ ton};$$

(*b*) $$0.81\times\frac{0.070}{0.095}=0.60 \text{ ton};$$

(*c*) $$0.81 \times \frac{0.055}{0.095} = 0.47 \text{ ton};$$

(*d*) $$0.81 \times \frac{0.046}{0.095} = 0.39 \text{ ton}.$$

Cost of Refrigeration per Square Foot per Year.

(*a*) $0.81 \times 40 = \$0.324$;

(*b*) $0.60 \times 40 = 0.240$;

(*c*) $0.47 \times 40 = 0.188$;

(*d*) $0.39 \times 40 = 0.156$.

Cost of Space Required per Year.

(*a*) $\frac{2\frac{1}{2}}{12} \times 1 \times 0.05 \times 12 = 0.120$;

(*b*) $\frac{3\frac{1}{2}}{12} \times 1 \times 0.05 \times 12 = 0.180$;

(*c*) $\frac{5}{12} \times 1 \times 0.05 \times 12 = 0.240$;

(*d*) $\frac{6}{12} \times 1 \times 0.05 \times 12 = 0.300$.

Yearly Cost of Insulation Investment per Year.

(*a*) $0.25 \times 0.15 = \$0.037$;

(*b*) $0.30 \times 0.15 = 0.045$;

(*c*) $0.40 \times 0.15 = 0.060$;

(*d*) $0.50 \times 0.15 = 0.075$.

Total Yearly Cost.

(*a*) $0.324 + 0.120 + 0.037 = \$0.481$;

(*b*) $= 0.456$;

(*c*) $= 0.488$;

(*d*) $= 0.521$.

From the above it is seen that for the assumed conditions the 3-in. thickness is the best. If conditions (assumed data) be changed, a different result will be obtained. If space is worth $2\frac{1}{2}$ cents per cu.ft. per month, a wall made up of two 2-in. boards would be best.

Problem 2. Find the **space required to store** the following: 180,000 doz. eggs, 25,000 bu. potatoes, 200,000 lbs. butter, 200,000 lbs. cheese, 25,000 bu. apples, 400,000 lbs. beef, 200,000 lbs. mutton, 200,000 lbs. pork, 20,000 lbs. poultry, 500 crates celery, 2000 bbls. vegetables, 2000 boxes oranges and lemons.

(*a*) *Size Egg Room.*

$$\text{No. of cases} = \frac{180{,}000}{30} = 6000;$$

$$\text{Volume of cases} = 6000 \times 2\tfrac{1}{4} = 13{,}500 \text{ cu.ft.};$$

$$\text{Height of piles} = 6 \text{ ft.};$$

$$\text{Net floor area} = \frac{13{,}500}{6} = 2250 \text{ sq.ft.};$$

Allow $\frac{1}{3}$ for aisles.

$$\text{Total floor area} = 2250 \times \tfrac{3}{2} = 3375 \text{ sq.ft.}$$

(*b*) *Size Potato Room.*

$$\text{No. of barrels} = \frac{25{,}000}{2.5} = 10{,}000 \text{ bbls.};$$

$$\text{Space required} = 10{,}000 \times 5 = 50{,}000 \text{ cu.ft.};$$

$$\text{Height of piles} = 8 \text{ ft.};$$

$$\text{Net floor area} = \frac{50{,}000}{8} = 6250 \text{ sq.ft.};$$

Allow $\frac{1}{5}$ for aisles.

$$\text{Total floor area} = 6250 \times \tfrac{5}{4} = 7812 \text{ sq.ft.}$$

(*c*) *Size Butter Room.*

$$\text{No. of tubs} = \frac{200{,}000}{50} = 4000;$$

$$\text{Space} = 4000 \times 2 = 8000;$$

$$\text{Height of piles} = 6 \text{ ft.}$$

Net floor area $= \frac{8000}{7} = 1143 = 1200$ sq.ft.;

Total floor area = 1400 sq.ft.

(*d*) *Size Cheese Room.*

Same as (*c*), 1400 sq. ft.

(*e*) *Size Apple Room.*

Same as (*b*), 7812 sq.ft.

(*f*) *Size of Beef Room.*

No. of halves of beef $= \frac{400{,}000}{750} \times 2 = 1070.$

Assume these to be hung at 18-in. intervals and 3 ft. apart.
Floor space $= 1070 \times 1\frac{1}{2} \times 3 = 4800$ sq.ft.

(*g*) *Size of Mutton Room.*

Assume carcass weighs 60 lbs per cu.ft. in piles and that piles are 4 ft. high with aisles occupying one-quarter space.

Floor space $= \frac{200{,}000}{60 \times 4} \times \frac{4}{3} = 1100$ sq.ft.

(*h*) *Size of Pork Room.*

Same as (*g*), 1100 sq.ft.

(*i*) *Size of Poultry Room.*

Same as (*g*), 1100 sq. ft.

(*j*) *Size of Celery Room.*

Total volume $= 500 \times 10 = 5000$ cu.ft.;
Height of piles = 8 ft.;

Net floor area $= \frac{5000}{8} = 625$ sq.ft.;

Allow $\frac{1}{3}$ for aisles.
Total floor area $= 625 \times \frac{3}{2} = 940$ sq.ft.

(*k*) *Size of Vegetable Room.*

Space = 2000 bbls. × 5 = 10,000 cu.ft.;

Height piles = 8 ft.;

$$\text{Net floor area} = \frac{10{,}000}{8} = 1250 \text{ sq.ft.};$$

Allow $\frac{1}{5}$ for aisles.

Total floor area = $1250 \times \frac{5}{4} = 1600$ sq.ft.

(*l*) *Size of Orange Room.*

Space = 2000 × 4 = 8000 cu.ft.;

Height = 5 ft.;

Net floor space = 1600 sq.ft.;

Allow $\frac{1}{5}$ for aisles.

Total floor area = 2000 sq.ft.

The layout shown in Fig. 185 is suggested for this problem. The height of the stories would be 10 ft. in the clear. The

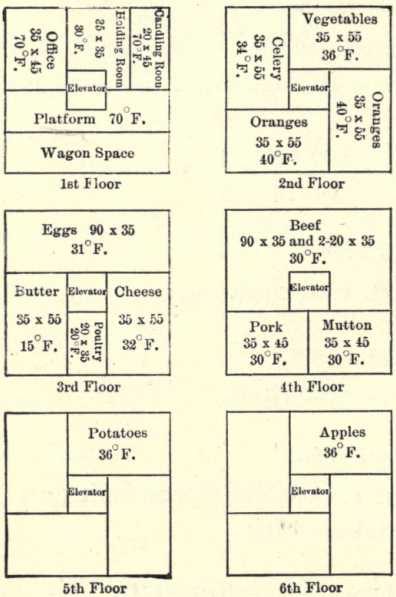


FIG. 185.—Typical Warehouse.

height of the building with a 4-ft. basement and a 5-ft. attic would give a total height of 75 ft.

Problem 3. Find the probable **amount of refrigeration for the plant** in Problem 2, assuming that all goods are received at 70° F. and that 90° F. is the warmest weather. Insulation: Main walls, 8-in. brick with two 3-in. thicknesses of cork attached with plaster and plaster finish. Partitions: 8-in. tile with plaster and 2 ins. of cork on one side, plastered. Floors at second story and ceiling of sixth floor: second figure, Fig. 108; other floors: brick arches with $K = 0.25$.

(*a*) *Temperature of Rooms.*

Eggs, 31° F.;	Meat, 30° F.;
Cheese, 32° F.;	Apples, potatoes, vegetables, 36° F.;
Butter, 15° F.;	Oranges, lemons, 40° F.;
Poultry, 20° F.;	Celery, 34° F.

(*b*) *K's for Walls.*

K for walls (from p. 211) $= 0.039$;
K for partitions.

$$K = \frac{1}{\frac{1}{0.21} + \underbrace{\frac{2}{12}}_{0.022} + \underbrace{\frac{1}{2 \times 12}}_{0.46}} = 0.081.$$

0.21 for tile with plaster, p. 211;
0.022 from p. 193 for compressed cork;
0.46 from p. 193 for plaster.

K for floors.

$K = 0.062$ p. 211 (second story);
$= 0.25$ assumed for hollow arch bricks without insulation (3d to 5th);
$= 0.030$ (p. 211), ceiling, top floor.

(*c*) *Heat Loss through Walls, Floors, Ceilings, Partitions.*

Heat loss from rooms in B.t.u. per hour.

First Floor

Holding room:		
Wall, $35\times10\times(90-30)\times0.039$	=	820
Partitions $(45+15+10+20+35)\times10\times(90-30)\times0.081$	=	6750
Floor	=	00
Ceiling, $35\times35\times(36-30)\times0.062$ $10\times15\times(40-30)\times0.062$	=	549
		8119
Total for first floor		8119

Second Floor

Celery room:		
Walls $(55+35)\times10\times(90-34)\times0.039$	=	1965
Partitions $[35\times10\times(36-34)+20\times10\times(90-34)+35\times10\times(40-34)]\times0.081$	=	1134
Floor $35\times55\times(90-34)\times0.062$	=	6684
Ceiling $[35\times35\times(31-34)+35\times20\times(15-34)]\times0.25$	=	−4244
		5539
Vegetable room:		
Walls $(55+35)\times10\times(90-36)\times0.039$	=	1895
Partitions $[35\times10\times(34-36)+20\times10(90-36)+35\times10(40-36)]\times0.081$	=	931
Floor $[35\times35\times(30-36)+20\times35(90-36)]\times0.062$	=	1888
Ceiling $(35\times55)\times(31-36)\times0.25$	=	−2406
		2308
Orange room I:		
Walls $(55+35)\times10\times(90-40)\times0.039$	=	1755
Partitions $[35\times10\times(36-40)+20\times10\times(90-40)+35\times10(40\times40)]\times0.081$	=	697
Floor $[(35\times45)(90-40)+10\times15(30-40)+20\times10\times(90-40)]\times0.062$	=	5410
Ceiling $35\times55\times(32-40)\times0.25$	=	−3850
		4012

Orange room II.

Walls $(55+35)\times 10\times(90-40)\times 0.039$	=	1755
Partitions $[35\times 10\times(34-40)+20\times 10\times(90-40)+35\times 10\times(40-40)]0.081$	=	640
Floor $35\times 55\times(90-40)\times 0.062$	=	5964
Ceiling $[35\times 35\times(15-40)+20\times 35\times(20-40)]\times 0.25$	=	−11,156
		−2797
Total for second floor		9062

THIRD FLOOR

(In the same manner as before.)

Egg room	6,760
Butter room	22,642
Cheese room	5,318
Poultry room	7,128
Total for third floor	41,848 B.t.u.

FOURTH FLOOR

Beef room	12,433
Mutton room	4466
Pork room	−742
Total for fourth floor	16,157 B.t.u.

FIFTH FLOOR

Total for floor ..... 22,632 B.t.u.

SIXTH FLOOR

Total for floor ..... 23,554 B.t.u.

Totals:

First floor	8,119
Second floor	9,062
Third floor	41,848
Fourth floor	16,157
Fifth floor	22,632
Sixth floor	23,554
	122,372

Tons Required: $\dfrac{122{,}372}{60\times 199.2} = 10.25.$

(*d*) *Heat from Goods.* The greatest amount received at one time will have to be assumed together with the time required to reduce the goods to warehouse conditions. The cooling is assumed to take place in forty-eight hours. The greatest amount received at any one time is assumed to be the following:

Beef, 40,000 lbs.;	Butter, 6000 lbs.;
Mutton, 15,000 lbs.;	Poultry, 2000 lbs.;
Pork, 25,000 lbs.;	Apples, 300 bbls.;
Eggs, 300 crates;	Potatoes, 200 bbls.

Heat Removed (see p. 215):

From Beef:	$40{,}000[(70-30)\times 0.70+90]$	= 4,720,000
Mutton:	$15{,}000[(70-30)\times 0.67+90]$	= 1,752,000
Pork:	$25{,}000[(70-30)\times 0.50+90]$	= 2,750,000
Eggs:	$(300\times 50)[(70-31)\times 0.76]$	= 444,600
Butter:	$6000[(70-15)\times 0.60+84]$	= 702,000
Poultry:	$2000[(70-20)\times 0.80+102]$	= 284,000
Apples:	$(300\times 150)[(70-36)\times 0.92]$	= 1,407,600
Potatoes:	$(200\times 150)[(70-36)\times 0.80]$	= 816,000
Total		12,876,200

Tons Required: $$\frac{12{,}876{,}200}{48\times 60\times 199.2}=22.4.$$

(*e*) *Heat from Lights* (p. 341).

In rooms 55×35 there would probably be ten 20-watt lamps. If six rooms are being used at one time the heat from these will be

$$T_e=\frac{10\times 20\times 6\times 3.41}{199.2\times 60}=0.34 \text{ ton.}$$

(*f*) *Heat from Men* (p. 214).

Assume 10 men working at one time.

$$T_m=\frac{10\times 1150}{199.2\times 60}=0.96 \text{ ton.}$$

(*g*) *Leakage through Doors.* This is a difficult quantity to estimate and if the assumption is made that $K=2$ and that the

door area is 8×8 and that 10 of these may be open at one time, the following results:

$$T_l = \frac{10 \times 64 \times (90-32) \times 2}{60 \times 199.2} = 6.2 \text{ tons.}$$

Totals:

Walls	10.25
Goods	22.40
Lights	.34
Men	.96
Leakage	6.20
	40.15

If 33% excess is allowed for safety the total will be 54 tons. **To check this,** various general rules will be used.

Volume of Building:

$$90 \times 90 \times 50 + (35 \times 35 + 150) \times 10 = 418{,}750 \text{ cu.ft.}$$

From **Peterson's rule,** on p. 256, using 20,000 cu.ft. as the average size of the room and 30° as the average temperature:

$$\text{Cu.ft. per ton} = 2.5 \times 1000 \left[\frac{20{,}000}{5000}\right]^{1/3} = 4000;$$

$$\text{Tons required} = \frac{418750}{4000} = 104 \text{ tons.}$$

From **average rule** on p. 256:

$$\text{Tons required} = \frac{418750}{3000} = 140 \text{ tons.}$$

The **difference in these results** is due to the assumptions made. In the first case the insulation is heavy and the time to cool goods is moderately long. If this time were made less and the insulation poorer, the tonnage would be increased. In the general rules, there is no specific value for these items.

In this problem two 30-ton machines should be installed.

Problem 4. Find the **amount of radiation to be placed in the room** for beef storage.

Heat from walls	=	12,433 B.t.u. per hr.
Heat from beef $=\frac{4{,}760{,}000}{48}$	=	100,000 B.t.u. per hr.
Heat from lights $= 20 \times 20 \times 3.41$	=	1364 B.t.u. per hr.
Heat from men $= 5 \times 1150$	=	5750 B.t.u. per hr.
Heat from door $= 64 \times 2 \times 60$	=	7680 B.t.u. per hr.
		127,227 B.t.u. per hr.

(*a*) *Direct Expansion.*

$t_r = 30°$ F.;
$t_a = 20°$ F.;
$K = 5$ (p. 244);

$$F = \frac{127{,}227}{10 \times 5} = 2544 \text{ sq.ft.}$$

Using extra heavy 2-in. pipe, 1.608 ft. of length will give 1 sq.ft.

Total Length $= 2544 \times 1.608 = 4100$ lin. ft.

From rules on pp. 255 and 256 the following is obtained:

Volume of room	45,500
Temperature	30° F.

Allow $\frac{2}{3}$ of 25 cu.ft. per lin. ft. on account of first freezing.

$$\frac{45{,}500}{\frac{2}{3} \times 25} = 2700 \text{ lin. ft.}$$

Allow $\frac{3}{2}$ of 35 sq.ft. per 1000 cu.ft. to allow for freezing in forty-eight hours.

$$\text{Surface} = \frac{45{,}500}{1000} \times \frac{3}{2} \times 35 = 2350 \text{ sq.ft.}$$

$$\text{Lin. ft.} = 2350 \times 1.608 = 3790 \text{ lin. ft.}$$

Amount allowed for room, 4000 lin. ft. This will be arranged in eight coils $37\frac{1}{2}$ ft. long and four coils 25 ft. long. Each coil will be five pipes high and two pipes wide.

(*b*) *Brine.* Surface, first method, using $7\frac{1}{2}°$ difference in place of 10° will require 6100 lin. ft.

Levey's table: $\text{Lin. ft.} = \frac{45{,}500}{\frac{2}{3} \times 17} = 4000$ lin. ft.;

Schmidt's table: $\text{Lin. ft.} = \frac{45{,}500}{1000} \times \frac{3}{2} \times 50 \times 1.6 = 5460$ lin. ft.

Amount to be used, 6000 lin. ft.

Problem 5. Find the **length of 2-in. brine pipe with a drop of 5°** and a mean temperature difference of $7\frac{1}{2}°$ F. between room and brine, assuring a 4-ft. per second velocity of brine.

Internal area—standard 2″ pipe	3.355 sq.in.
Outside circumference—standard 2″ pipe	7.461 in.
Brine, sp. gr	1.119
Specific heat	0.8
K	5.0

From (19) p. 259:

$$L = \frac{\frac{3.355}{144} \times 4 \times 3600 \times 62.4 \times 1.119 \times 0.8 \times 5}{\frac{7.46}{12} \times 7\frac{1}{2} \times 5} = 4260 \text{ ft.}$$

In table on page 255 it is noted that Levey suggests that these coils be made 275 ft. long. If such is done there will be different conditions from those noted: First, the velocity must be much less than 4 ft., and second the temperature drop will be less than 5° F. Of course, if K is taken as 10 instead of 5, there will be a decrease in length. If 1 ft. per second is used as velocity, and the drop is taken as 2°, although the temperature drop is $7\frac{1}{2}°$, and if K is used as 10, the length is found to be about 215 ft.

$$L = 4260 \times \tfrac{5}{10} \times \tfrac{1}{4} \times \tfrac{2}{5} = 213 \text{ ft.}$$

Problem 6. Find the **velocity of brine in a 2-in. coil,** 190 ft. long, if the mean temperature difference is 5° F., $K=5$, and temperature drop is 5° F.

$$Q=190\times\frac{7.461}{12}\times5\times5=\frac{3.355}{144}\times w\times3600\times62.4\times1.119\times0.8\times5$$

$$w=0.127 \text{ ft. per sec.}$$

Problem 7. Find the amount of **ammonia** which will be **evaporated** in a 2-in. coil, 190 ft. long, of extra heavy pipe, if the ammonia is at 60° F. before throttling it to 20° F., and the temperature difference is 10° F.

$$\textit{Heat from Pipe}=190\times\frac{7.461}{12}\times5\times10=5900 \text{ B.t.u. per hr.}$$

$$\textit{Heat for 1 lb. of Ammonia}=(i_1-i_5)=512.8(75) \text{ p. 69.}$$

i for 60° F. and $x=0=30.9$

i for 20° F. and $x=1=543.7$

$$\text{Pounds of ammonia per hour}=\frac{5900}{512.8}=11.5 \text{ lbs.}$$

Quality of ammonia after throttling to

$$20°,\ x=\frac{30.9-(-12.6)}{556.3}=0.078.$$

Specific volume after throttling

$$=0.078\times5.92+0.922\times0.0244=0.484.$$

Velocity of mixture in 2-in. pipe at entrance:

$$w=\frac{11.93\times0.48}{2.953\times3600}\times144=0.078 \text{ ft. per sec.}$$

If longer pipes are used a greater quantity will be admitted and a higher velocity will be used.

Problem 8. Find the **amount of air** to be admitted **in an indirect system** for the data of Problem 4 during the time of filling. Find the **surface required in bunker coils.**

Heat removed per hour = 127,227 B.t.u.;
Temperature of room, 30° F.;
Assumed temperature of air, 20° F.

$$\textit{Amount of air per minute} = \frac{127{,}227}{10 \times 0.02 \times 60} = 10{,}602 \text{ cu.ft.}$$

This assumes that air retains the same amount of moisture.

Use 1-in. pipes for bunker coils. Assume velocity of air 900 ft. per minute.

Area through clear space in bunker

$$= \frac{10{,}602}{900} \times 144 = 1700 \text{ sq.in.}$$

If pipes are 6 ft. long and 1 in. is allowed between pipes, the number of sections will be

$$\frac{1700}{72} = 24.$$

$$K = 1 + 1.3\sqrt{15} = 6.03, \text{ (14), p. 256.}$$

Air is cooled from 30 to 20 with ammonia at 15° F.

$$\text{Mean } \Delta t = \frac{15 - 5}{\log_e \frac{15}{5}} = 9.1.$$

$$\text{Surface} = \frac{127{,}227}{9.1 \times 6.03} = 2320 \text{ sq. ft.}$$

or $2320 \times 2.904 = 6750$ lin. ft.

$$\text{Lines of pipe per section} = \frac{6750}{24 \times 6} = 47.$$

This excessive number of lines and surface is due to the small difference of temperature assumed. If a greater difference in temperature were used, the coil surface would be smaller, but the cost of compression would be greater, as a lower back pressure would be needed.

This problem has not considered any change in moisture content. If this were taken into consideration, more surface would be required. The problem of heat and surface required

when there is a change of moisture content is given in Problem 25.

Problem 9. Find the **size of ducts** for air of Problem 8 together with **pressure drop, size of fan,** and **power** required.

Velocities in System (p. 248):

In register.....300 ft. per min.
In branches.....800 ft. per min.
In main.....1200 ft. per min.

$$\textit{Size of main} = \frac{10{,}602}{1200} = 8.8 \text{ sq.ft., } 4' \times 2.2'.$$

Assume main 80 ft. long with 5 bends, $4' \times 2.2'$.
Assume branch 20 ft. long with 3 bends, $2' \times 1'$.

$$\text{Loss in grill} = 0.8 \times \frac{(\frac{300}{60})^2}{64.3} = 0.31 \text{ (p. 250).}$$

$$\text{Loss in branch} = 0.02 \times \frac{20}{4 \times 0.33} \times \frac{(\frac{800}{60})^2}{64.3} = 0.83 \text{ (p. 249).}$$

$$R_1 = \frac{2 \times 1}{2(2+1)} = 0.33.$$

$$\text{Loss in 3 bends} = 3 \times 0.15 \times \frac{(\frac{800}{60})^2}{64.3} = 1.24.$$

$$\text{Loss in main} = 0.02 \times \frac{80}{4 \times 0.71} \frac{(\frac{1200}{60})^2}{64.3} = 3.51.$$

$$R_1 = \frac{8.8}{2 \times 6.2} = 0.71.$$

$$\text{Loss in 5 bends} = 0.15 \times 5 \times \frac{(\frac{1200}{60})^2}{64.3} = 4.66$$

$$\text{Loss in 47 lines of pipe} = 47 \times 0.4 \times \frac{(\frac{900}{60})^2}{64.3} = 65.6$$

$$\text{Velocity head at end} = \frac{(\frac{300}{60})^2}{64.3} = 0.39$$

Total loss 76.54 ft. of air.

Oz. pressure $= \frac{76.54}{110} = 0.696$ (p. 250).

Inches of water $= 0.696 \times 1.73 = 1.20$.

Dynamic pressure for Sirocco fan (p. 251) $= \frac{0.696}{0.712} = 0.977$ oz.

Equivalent tabular volume for 1 oz. $= 10{,}602\sqrt{\frac{1.00}{0.977}} = 10{,}750$.

No. 6 fan is the nearest fan in the table.

Speed $= 381 \times \sqrt{\frac{0.977}{1.00}} = 376$.

Discharge $= 11{,}300 \times \sqrt{\frac{0.977}{1.000}} = 11{,}150$.

Power $= 5.30 \times \frac{0.977}{1.000} \times \sqrt{\frac{0.977}{1.000}} = 5.12$.

This fan is slightly too large, but using tabular values only it is the one which must be selected.

Problem 10. Find the **size brine main, size of pump,** and **power** to pump brine for warehouse.

For 60 tons capacity and 10° drop in brine, the **weight of brine** to be circulated per minute is given by (18) p. 259.

$$M_b \times 0.8 \times 10 = 60 \times 199.2.$$

$$M_b = 1498 \text{ lbs.}$$

$$\text{Volume per min.} = \frac{1498}{62.4 \times 1.119} = 21.44 \text{ cu.ft.} = 160 \text{ gal. per min.}$$

$$\text{Area main} = \frac{21.4}{60 \times 4} \times 144 = 12.9 \text{ sq.in.}$$

Use 4-in. pipe. Area, 12.65 sq.in. This gives a velocity of 4.1 ft. per sec.

Duplex pump size to discharge brine at 45 cycles per minute.

Assume 8-in. stroke.

$$\text{Area piston} = \frac{21.44 \times 1728}{45 \times 4 \times 8} = 25.9 \text{ sq.in.}$$

Diameter, $5\frac{3}{4}$ in.

Use 6×8-in. brine end to pump.

To find **power to drive** brine through warehouse it would be necessary to lay out all lines, branch circuits and compute losses in various parts. To find the approximate power it will be assumed that the 4-in. line extends to the top of the building and back again with 200 feet of pipe and eight right-angle bends, and the longest branch, near end, is 400 feet of 2-in. pipe with thirty right-angle bends. The velocity is 0.127 ft. per second in the branch and 4.1 ft. per second in the main. The main has branches taken off from it at intervals and is, therefore, equivalent to a main of length equal to one-third of the length on line.

From p. 260:

$$f = \frac{0.025}{(0.33 \times 4.1)^{\frac{1}{4}}} = 0.023$$

$$f = \frac{0.025}{(0.17 \times 0.127)^{\frac{1}{4}}} = 0.065.$$

$$Loss = 0.023 \times \frac{\frac{200}{3}}{0.33} \times \frac{(4.1)^2}{64.3} + 8 \times 0.2 \times \frac{\frac{8}{12}}{0.33} \times \frac{(4.1)^2}{64.3} + 0.065 \times \frac{400}{0.167}$$

$$\times \frac{(0.127)^2}{64.3} + 30 \times 0.2 \times \frac{5}{2} \times \frac{(0.127)^2}{64.2} = 2.27 \text{ ft.}$$

There is no head lost in forcing the brine to top of the system, as the pipe is full. The slight difference in density, due to 10° difference in temperature in ascending and descending pipes, is neglected.

Total hydraulic work $= 2.27 \times 1498 = 3410$ ft.lbs. per min.

Assuming 60% for the mechanical efficiency of pump, the power required to drive pump is

$$\text{I.H.P.} = \frac{3410 \times \frac{1}{.60}}{33{,}000} = 0.172 \text{ H.P.}$$

Problem 11. Find the **size of supply main for liquid ammonia** and **return main for vapor** for warehouse.

$$\text{Total ammonia per min.} = \frac{60 \times 199.2}{512.8} = 23.3 \text{ lbs. (See Prob. 7.)}$$

Volume of liquid at 60° F. $= 23.3 \times 0.02609 = 0.609$ cu.ft.

$$\text{Area pipe} = \frac{0.609}{4 \times 60} \times 144 = 0.365 \text{ sq.in.}$$

Diameter $= \frac{3}{4}''$.

Use 1″ extra heavy pipe.

Volume of vapor at 15° $= 23.3 \times 6.583 = 153$ cu.ft.

(*a*) *From Velocity Considerations.*

Assume velocity 60 ft. per sec. (p. 257).

$$\text{Area} = \frac{153}{60 \times 60} \times 144 = 6.1 \text{ sq.in.}$$

Use extra heavy 3-in. pipe.

(*b*) *From Pressure Drop Considerations.*

Assume $\frac{1}{4}$ lb. drop in 50 ft. of pipe (p. 257).

$$\frac{1}{4} = 0.00015\left[1 + \frac{3.6}{d}\right]\frac{(24.35)^2 \times 50}{0.1519 \times d^5}.$$

Use 3″ for d within bracket.

$d^5 = 258$.

$d = 3.16$.

Use $3\frac{1}{2}''$ extra heavy pipe.

Problem 12. Find the **size of a freezing tank** for a 50-ton plant using 300-lb. cans.

Size of can from p. 290: $11\frac{1}{2} \times 22\frac{1}{2}$ at top, $10\frac{1}{2} \times 21\frac{1}{2}$ at bottom, 45 in. length over all, 44 in. length inside.

Surface transmitting heat for 42 in. water depth

$$= \{2[11 \times 42] + 2(22 \times 42) + 10\tfrac{1}{2} \times 21\tfrac{1}{2}\}\tfrac{1}{144} = 20.83 \text{ sq.ft.}$$

Heat per hour with 20° brine and a coefficient of 3.3 B.t.u.

$$= 20.83 \times (32 - 20) \times 3.3 = 826.$$

Heat from can $= 300[144.3 + (40 - 32)] = 48{,}390$ B.t.u.
(Temperature of water from cooler, 40°; p. 308.)
(Temperature of ice 20° F.)

$$\textit{Hours to remove heat} = \frac{48{,}390}{826} = 58.5.$$

By (3) p. 304:

$$\textit{Time to freeze} = \frac{4.2 \times \overline{11.5}^{2}}{32 - 20} = 46.3 \text{ hrs.}$$

These do not check, because the value of K used above is lower than can be used. If 4 were used in place of 3.3, the two would check. However, the previous value of 3.3 will be used. If now the temperature of the brine is reduced to 16° F. the heat removed per hour would be

$$20.83 \times (32 - 16) \times 3.3 = 1100.$$

$$\text{Hours} = \frac{48{,}390}{1100} = 44 \text{ hrs.}$$

Total **cans per ton** per day with 15% allowance

$$= \frac{2000}{24 \times 300} \times 44 \times 1.15 = 14.1.$$

From p. 305 it is seen that 14 cans are allowed per ton, but 16 cans are allowed by Shipley (p. 307). Using 16, the total number of cans would be

Number of cans $= 50 \times 16 = 800$ cans.

If this is made 16 cans wide and 50 cans long the tank sizes will be

$$\text{Length} = 50[11\tfrac{1}{2} + 1 + 1\tfrac{1}{2} + 1] = 62' \; 6'';$$

$$\text{Width} = 16[22\tfrac{1}{2} + 1 + 3 + 1] = 36' \; 8'';$$

$$\text{Depth} = 45'' + 6'' = 4' \; 3''.$$

Problem 13. Find the **space required for a plate plant** of 50 tons capacity, using ammonia at 16° F.

Volume of ice per day $= \frac{50 \times 2000}{57.5} = 1740$ cu.ft.

Total length if 8-ft. depth and 12-in. thickness is used

$$= \frac{1740}{1 \times 8} = 220 \text{ ft.}$$

If 6 coils are used, giving 12 plates, the length is given by

$$\frac{220}{12} = 18\tfrac{1}{3} \text{ ft.}$$

Using 12-in. clearance the length of the tank will be

$$12 \times 12'' + 6 \times 12'' + 6'' + 6'' = 19'.$$

Tank size $= 19' \times 18\tfrac{1}{3}' \times 3'$.

Time to freeze $= \frac{21 \times \overline{12}^2}{32 - 16} = 189$ hrs. $= 7.9$ days $= 8$ days (p. 304).

Eight tanks will be required.

Floor space $= 43' \times 92'$.

Problem 14. Find the **coil surface required for the tank** of Problem 12.

Heat removed per pound of ice made $= 200$ B.t.u. (p. 308)

Heat from coil $= \frac{50 \times 2000 \times 200}{24} = 833{,}400$ B.t.u. per hr.

Assume 10° difference between ammonia and brine.

$K = 15$ (p. 306).

Surface of coil $= \frac{833{,}400}{10 \times 15} = 5560$ sq.ft.

Linear feet of $1\frac{1}{4}$-in. pipe $= 5560 \times 2.301 = 12{,}800$ ft.

From p. 307 the requirement would be

$$\text{Lin. ft.} = 250 \times 50 = 12{,}500.$$

If 17 coils 60 ft. long are used the coils must be 13 pipes high.

$$\text{Number of pipes} = \frac{12{,}800}{60 \times 17} = 12.5.$$

This requires 41 in. of height if 3 centers are used.

Problem 15. Find the **tons of refrigeration for plant** of Problem 12, if 30 B.t.u. are required for cooling.

$$\text{Tons of refrigeration} = \frac{50 \times 2000 \times (200+30)}{24 \times 60 \times 199.2} = 80.2.$$

Problem 16. Find whether or not **ice storage will pay** with load curve shown on p. 309.

Average machine capacity, 150-tons.

Peak load capacity, 325 tons.

Ice to be stored with 150 ton machine:

May	$25 \times 31 =$	775
June	$65 \times 30 =$	1950
July	$175 \times 31 =$	5425
August	$170 \times 31 =$	5270
September	$130 \times 30 =$	3900
October	$55 \times 31 =$	1705
Total		19,025 tons.

$$\text{Volume of ice} = \frac{19{,}025 \times 2000}{57.5} = 662{,}000 \text{ cu.ft.}$$

Size of house $= 130 \times 130 \times 40 = 676{,}000$ cu.ft.

Cost of building $= 0.06 \times 676{,}000 = \$40{,}560.00.$

(From p. 344.)

Insulation, two 2″ cork. $0.40 \times 37{,}700 =$ 15,080.00.

$55,640.00

Yearly cost on building:

Interest	6%
Taxes and Insurance	1%
Depreciation	5%
Repairs	1%
	13%

13% of \$55,640.00 = \$ 7,233.20

Cost of handling and holding $= \$0.25 \times 19{,}025 =$ 4,756.25

(p. 375.)

Total cost of storing ice \$11,989.45

Cost of extra apparatus if no storage is used:

175-ton compressor and engine with condenser, piping, receiver, oil separator	$ 23,000.00
Boilers $\left(\frac{175\times 2.5\times 20}{30} = 292 \text{ Boiler H.P.}\right)$, including chimney piping, pump (p. 378)	9,000.00
Cans, tank and coils	19,000.00
Distilling apparatus	3,800.00
Erection	9,000.00
Additional building space 400,000 at 10 cts. (100×20×20)	40,000.00
Total cost	$102,800.00

From p. 347, the cost per ton is $500.00, giving as the probable cost	$ 87,500.00
Amount assumed as cost	95,000.00

With 8% depreciation and 3% for repairs the fixed charges will be 18%.

Fixed cost per year = 18% × 95,000	= $17,100.00
Additional labor cost	1,500.00
Total	$18,600.00
Saving by use of storehouse = $18,600 − $12,000	= $6,600.00

Problem 17. Find the **cost per ton of pumping water** for ice in a raw-water plant of 100 tons capacity if the water is 200 ft. below surface, using anthracite buckwheat coal.

$$\text{Water per minute} = \frac{100\times 2000\times 1.15}{24\times 60} = 160 \text{ lbs.}$$

$$\text{Water horse-power} = \frac{160\times 200}{33{,}000} = 0.965.$$

(*a*) *Power of steam end of deep-well pump* $= \frac{0.965}{0.75} = 1.29$ H.P.

Steam required = 1.29 × 120 = 155 lb.
(p. 351.)

Pounds coal per hour $= \frac{155 \times 1100}{12{,}800 \times 0.75} = 16.1$

Tons of coal for water pumping per ton ice $= \frac{16.1 \times 24}{100 \times 2240}$
$= 0.00172$ ton.

Cost of coal for pumping per ton of ice $= 0.00172 \times \$3.10$
$= \$0.00533$.

Cost of attendance per ton of ice (50 cts. per ton of coal)
$= \$0.0008$.

Cost of pump end equivalent part of boiler, assumed $\$100.00$.

At 20%, fixed charges per ton of ice $= \frac{\$100.00 \times 0.20}{365 \times 100}$
$= \$0.00055$.

Total cost of pumping per ton of ice $= \$0.00668$.

(*b*) *Power for air-lift pump* $= \frac{0.965}{0.40} = 2.41$ H.P.

Pounds of steam per hour $= 2.41 \times 35 = 84.4$ lbs.

Tons of coal per ton of ice $= \frac{84.4 \times 1000 \times 24}{12{,}800 \times 0.75 \times 100 \times 2240}$
$= 0.00095$.

Cost of coal for pumping water per ton of ice $= 0.00095 \times \$3.10 = \$0.00244$.

Cost of attendance, $= \$0.00048$.

Cost of pump end equivalent part of boiler, $= \$250$.

Cost per ton of ice for fixed charges $= \frac{\$250 \times 0.20}{100 \times 365}$
$= \$0.00137$.

Total cost of water pumping per ton of ice $= \$0.00429$.

Problem 18. Find the steam and surface necessary to evaporate 40 tons of water per 24 hours, with steam at 5.3 lbs. per sq.in. gauge pressure and of quality 1.00.

Temperature of steam at 20 lbs. abs.	228° F.
Temperature assumed in evaporator	193° F.
Pressure in evaporator, lbs. abs.	9.96
Vacuum in evaporator	9.7″
Heat content of steam entering	1157.7
Heat content of steam leaving	1144.3
Heat content of water at 193° F	160.92
External surface of evaporator, assumed	250 sq.ft.

Heat loss from 2 in. of 85% magnesia $= 250 \times \frac{0.035}{\frac{2}{12}}[193 - 90]$

$= 5408$ B.t.u., using (1) p. 302.

$$M_s(1157.7 - 160.92) = \frac{40 \times 2000}{24}[1144.3 - 160.92] + 5408$$

$$= 3{,}278{,}000 + 5408 = 3{,}283{,}408.$$

$$M_s = \frac{3{,}283{,}408}{996.7} = 3300 \text{ lbs. per hr.}$$

$$\text{Tube surface required} = \frac{\frac{40 \times 2000}{24}(1144.3 - 160.9)}{400(228 - 193)} = 4205 \text{ sq.ft.}$$

This requires thirty-two 4-in. tubes 6 ft. long, using a drum about 5 ft. in diameter.

Problem 19. Find the **size of filter** to be used for filtering the raw water for a 100-ton plant.

$$\text{Water} = \frac{100 \times 2000 \times 1728}{62.5 \times 24 \times 231 \times 60} \times 1.15 = 19.1 \text{ gal. per min.}$$

Allow 2.5 gal. per min, per sq.ft. of filter (p. 287).

$$\text{Area of deck} = \frac{19.1}{2.5} = 7.6 \text{ sq.ft.} = 1094 \text{ sq.in.} \quad \text{Diam.} = 38''.$$

Problem 20. Find the **size of compressor** to be used for a raw-water ice plant of 100 tons capacity.

(*a*) *Find Power.*

100 tons requires 1600 cans of 300-lb. capacity each.

Air required $= \frac{1}{3}(1.8 + 0.3) = 0.7$ cu.ft. per min. per can.

Total air $= 0.7 \times 1600 = 1120$ cu.ft. of free air per min.

Assume 90% for volumetric efficiency of a compressor running at 120 R.P.M.

$$\text{Displacement} = \frac{1120}{0.90 \times 240} = 5.2 \text{ cu.ft.} = 9000 \text{ cu.in.}$$

Use 20″×30″. Displacement = 9560 cu.in.

Power required = 0.4 × 100 = 40.0 H.P. (p. 296).

Power required for air compressor by calculations to compress 1120 cu.ft. per minute to 18 lbs. per sq.in. by gauge will be found to be 64 H.P.

Problem 21. Find the **power to drive the compressor** required for Problem 15, the **size of the compressor and parts,** the **amount of water** for the condenser, and the **condenser surface.** Temperature of cooling water 65° F. (See pp. 72, et seq.)

Temperature of evaporation	6° F.
Temperature of cooling water at inlet	65° F.
Temperature of water at point at which ammonia reaches the saturated state	80° F.
Temperature of after-cooled liquid	75° F.
Temperature of ammonia in saturated portion of condenser	90° F.
Heat content of dry saturated ammonia at 6° F	540.1
Entropy of dry saturated ammonia at 6° F.	1.1616
Specific volume of dry saturated ammonia at 6° F	8.02
Pressure at 6° F	34.60 lbs. per sq.in.
Pressure at 90° F	181.80 lbs. per sq.in.
Heat content at 181.8 and entropy 1.1616.	644.0 B.t.u.
Heat content of dry vapor at 181.8	558.9 B.t.u.
Heat content of liquid at 90° F	65.3 B.t.u.
Heat content of liquid at 75° F	47.8 B.t.u.
Temperature of ammonia at end of compression	211° F.
Specific volume at end of compression	2.18 cu.ft.

Amount of ammonia to produce 80.2 tons

$$= \frac{80.2 \times 199.2}{540.1 - 47.8} = 32.5 \text{ lbs. per min.}$$

Power to drive compressor $= \frac{32.5(644.0 - 540.1)}{42.42 \times 0.75} = 102$ H.P.

The **size of the compressor** is fixed after the clearance is known. The **clearance** is made small so as to give a large volumetric efficiency. The clearances of $\frac{1}{16}$ in. on the upper end and $\frac{1}{8}$ in. on the bottom end have been used on double-acting cylinders and in single-acting cylinders with a safety head $\frac{1}{32}$ in. and even $\frac{1}{64}$ in. have been used on the upper end, while that at the lower end may be anything. With small clearances the **clearance volume** will amount to about $\frac{1}{2}\%$.

As was pointed out earlier there is no **effect of clearance on the work** done, except in a slight degree, due to friction from longer strokes with larger clearances. The **effect on volumetric efficiency** is quite marked and hence, the amount of ammonia handled at a given speed and with it the amount of refrigeration. The **York Mfg. Co.** has performed a number of **experiments** on their double- and single-acting compressors with various amounts of clearance and has obtained the results given in the following table:

COMPRESSOR I. H. P. PER TON FOR SINGLE-ACTING AND DOUBLE-ACTING COMPRESSORS WITH VARIOUS CLEARANCES

Linear Clearance	Clearance Volume in % of Displacement		H.P. at 5 Lbs. Suction Pressure		H.P. at 15.67 Lbs. Suction Pressure		H.P. at 25 Lbs. Suction Pressure	
	S.A.	D.A.	S.A.	D.A.	S.A.	D.A.	S.A.	D.A.
$\frac{1}{32}''$	0.24	.....	1.75	.....	1.30	.....	1.09	
$\frac{3}{64}''$	.....	0.42	.....	2.18	.....	1.60	.....	1.26
$\frac{1}{8}''$	0.76	0.85	1.77	2.34	1.32	1.62	1.10	1.28
$\frac{1}{4}''$	1.46	1.55	1.81	2.45	1.34	1.64	1.11	1.30
$\frac{1}{2}''$	2.85	2.93	1.82	2.56	1.36	1.72	1.12	1.35
1''	5.63	5.71	1.83	2.89	1.39	2.01	1.13	1.44

NOTE.—S. A. Single-acting Compressor.
D. A. Double-acting Compressor.
Clearance Volume includes indicator connections, valve shut.

TONNAGE PER 24 HOURS FOR SINGLE-ACTING AND DOUBLE-ACTING COMPRESSORS WITH VARIOUS CLEARANCES

Linear Clearance	Clearance Volume % of Displacement		Tons at 5 Lbs. Suction Pressure		Tons at 15.67 Lbs. Suction Pressure		Tons at 25 Lbs. Suction Pressure	
	S.A.	D.A.	S.A.	D.A.	S.A.	D.A.	S.A.	D.A.
$\frac{1}{32}''$	0.24	.....	22.7	.....	38.0	.....	50.4	
$\frac{3}{64}''$	.....	0.42	.....	19.2	.....	33.0	.....	47.4
$\frac{1}{8}''$	0.76	0.85	22.6	17.3	37.2	32.1	50.1	45.1
$\frac{1}{4}''$	1.46	1.55	21.0	16.0	35.6	30.0	49.1	44.8
$\frac{1}{2}''$	2.85	2.93	19.7	14.3	34.4	28.9	47.0	42.3
$1''$	5.63	5.71	15.5	10.6	29.7	22.9	42.6	36.5

For $\frac{1}{2}\%$ clearance the **clearance factor** is

$$1+0.005-0.005\left(\frac{181.8}{34.6}\right)^{\frac{1}{1.33}}=0.988$$

If 5% **leakage** around valves and piston is assumed the **volumetric efficiency** is

$$0.95\times 0.988=0.939.$$

$$\textit{Displacement per min.}=\frac{32.5\times 8.02}{0.939}=278 \text{ cu.ft.}$$

The **piston speed** used in refrigeration work varies from 140 ft. per minute in compressors of 5 tons to 500 ft. per minute with compressors of 300 tons capacity. These give 140 R. P. M. for the small compressors and 50 R. P. M. for the large ones. Using 2 compressors with 2 cylinders each at 80 R. P. M. the displacement of one cylinder is

$$\text{Displacement}=\frac{278}{2\times 2\times 80}=0.87 \text{ cu.ft.}=1500 \text{ cu.in.}$$

Sizes 10×19.
12×13.

Either of the above might be used as the ratio of stroke to diameter used in practice varies from 2 to 1 to 1 to 1.

The **cylinders** are made of close-grain cast iron. They are designed to stand 300 per sq.in., using the ordinary formulæ for

cylinder thickness. One-quarter to $\frac{3}{8}$ in. is added for reboring. The upper portion of the cylinder is jacketed.

Valves. The valves should be as large as possible. About 25% of the piston area is sometimes found in valve area. The **velocity through the valves** may be used to determine the valve area. In this case 4000 ft. per minute is used in the suction valves where the increase of pressure is noticeable, while 10,000 ft. per minute has been used through discharge valves. In any case make the area as large as possible. In single-acting compressors or their equivalent the suction valves are practically as large as the piston. The discharge valves may be single large valves as in the Frick vertical or a number of smaller valves as in the Frick horizontal compressor.

Pipe Connections. The pipe connections are of such a size that the velocity is 4000 ft. per minute on the suction side and 8000 on the discharge.

$$F_{suc} = \frac{32.5 \times 8.02}{2 \times 4000} \times 144 = 4.7 \text{ sq.in. or 3-in. extra heavy pipe.}$$

$$F_{dis} = \frac{32.5 \times 2.18}{2 \times 8000} \times 144 = 0.64 \text{ sq.in. or 1-in. extra heavy pipe.}$$

Use 3-in. and 2-in. pipes.

Pistons. The ammonia pistons are designed for 300 lbs. per sq.in. They are made deep. The depth is about equal to the diameter or $\frac{3}{4}$ the diameter of the piston. Some makers use three piston rings, ground to fit the ring groove, while others use four or five rings. The usual design of rings is made. The pistons are made of cast iron or steel. They are designed as flat plates supported by a series of beams. Empirical constants are found in handbooks of machine design.

Piston Rod. The piston rod should be made of high-grade alloy steel and a factor of safety of 10 should be used. The rod is attached to the piston by a thread, using the piston as a nut or using a separate nut. The section at the root of the thread should be designed for tension. The main body of the rod is designed as a column.

Condenser. *Heat removed in superheater portion* per pound of ammonia = 644.0 − 558.9 = 85.1 B.t.u.

Heat removed in saturated portion per pound of ammonia = 558.9 − 65.3 = 493.6 B.t.u.

Heat removed in after cooler per pound of ammonia = 65.3 − 47.8 = 17.5 B.t.u.

Amount of cooling water per minute

$$= \frac{32.5 \times [493.6 + 17.5]}{48.05 - 33.08} = 1110 \text{ lbs.}$$

$$= 17.75 \text{ cu.ft. per min.}$$

$$= 132.5 \text{ gal. per min.}$$

$48.05 = q'$ at 80 for water;

$33.08 = q'$ at 65 for water.

Temperature of water at end of superheat is given by

$$q_0 - q'_i = \frac{M_a(i_2' - i_2)}{M_w}.$$

$$q_0 = 48.05 + \frac{32.5 \times 85.1}{1110}$$

$$= 48.05 + 2.5 = 50.55.$$

$$t = 82.5.$$

Temperature of water at end of aftercooler and entrance to saturated portion is given by

$$q_2 - q_i' = \frac{M_a(q_5 - q_5')}{M_w};$$

$$q_2 = 33.08 + \frac{32.5 \times 17.5}{1110}$$

$$= 33.08 + 0.51 = 33.59;$$

$$t = 65.5.$$

Temperature Differences:

At inlet aftercooler.....	75 − 65	= 10
At outlet aftercooler.....	90 − 65.5	= 24.5
At inlet superheater.....	90 − 80	= 10
At outlet superheater.....	211 − 82.5	= 128.5

Mean Temperature Differences:

For aftercooler $\Delta T = \dfrac{24.5-10}{2.3 \log \dfrac{24.5}{10}} = 16.2;$

For saturated portion $\Delta T = \dfrac{24.5-10}{2.3 \log \dfrac{24.5}{10}} = 16.2;$

For superheated portion $\Delta T = \dfrac{128.5-10}{2.3 \log \dfrac{128.5}{10}} = 46.3.$

If the water is forced through the double-pipe condenser at 5 ft. per second, the **value of** K would be 275, from Fig. 94. From formula (18) the value is 291 and by (20) it is 220 with 2-in. and 3-in. pipes. On Fig. 188 the value of K is 100. The value of 200 will be used in the problem.

$$F_{sat} = \frac{60 \times 32.5 \times 493.6}{16.2 \times 200} = 297 \text{ sq.ft.}$$

For the aftercooler 400 will be used for K (p. 187).

$$F_a = \frac{60 \times 32.5 \times 17.5}{16.2 \times 400} = 5.27 \text{ sq.ft.}$$

For the superheated portion (22) p. 188 gives $K = 30$.

$$F_{sup} = \frac{60 \times 32.5 \times 85.1}{46.3 \times 30} = 122.0 \text{ sq.ft.}$$

Total surface with $\frac{1}{3}$ *increase* as safety factor = 566 sq.ft.

The **ordinary rules** call for from 8 to 18 sq.ft. per ton. This would give 640 to 1440 sq.ft. The difference between 556 and 640 is due to temperature difference. With water at temperatures taken 556 sq.ft. is sufficient.

If the condenser is made up of 2-in. and 3-in. pipes and no allowance is made for cooling from the outside, the **total length** will be

Total length = $566 \times 1.608 = 910$ ft.

Number of stands of 12 pipes, 20 ft. long = $\frac{910}{240} = 4$.

In **Block condensers** 9 lin.ft. per ton is allowed. This requires about 720 linear feet. **Shipley** uses 8 ft. per ton in his improved condenser. Ordinarily with temperatures occurring in practice 25 lin.ft. per ton may be allowed in double-pipe condensers of 2-in. and 3-in. pipes. In the problem just worked out about 11 lin.ft. per ton is used. This is due to the velocity of the water and the assumed temperatures.

Problem 22. If one-third of the refrigeration of Problem 21 is possible at 20° F. in place of 6° F., find the size of compressor and power required for this if a **Voorhees multiple effect installation** is used.

Refrigeration at 6° F. ..... 53.5 tons
Refrigeration at 20° F. ..... 26.7 tons

Dry compression or $x = 1$ at discharge from coils:

$i''_{6°}$ ..... 540.1 B.t.u.
$i''_{20°}$ ..... 543.7 B.t.u.
$i'_{75°}$ ..... 47.8 B.t.u.
v''_{6} ..... 8.02 cu.ft.
v''_{20} ..... 5.92 cu.ft.
$p_{6°}$ 34.6
$p_{20°}$ 47.75

M_1 ..... $\dfrac{53.5 \times 199.2}{540.1 - 47.8} = 21.7$

M_2 ..... $\dfrac{26.7 \times 199.2}{543.7 - 47.8} = 10.8$

Volume of cylinder for 21.7 lbs. at $6° = 21.7 \times 8.02 = 174$ cu.ft.

Volume of 1 lb. after adiabatic compression from 34.6 to 47.75, 6.32 cu.ft.

Volume of cylinder to care for addition at $20° = 21.7 \times 6.32 + 10.8 \times 5.92 = 201$ cu.ft.

If the compressor is built of 174 cu.ft. capacity, the 10.8 lbs. of ammonia will not be drawn in at 47.75 lbs. pressure and the refrigeration cannot be done while a displacement of 201 cu.ft.

per minute would lower the back pressure in the lower system to 29.3 lbs. per sq.in. and more work would have to be done.

$$\text{Specific volume} = \frac{201}{21.7} = 9.28;$$

$$\text{Pressure} \quad = 29.3.$$

This cannot be changed if it is necessary to divide refrigeration, as stated.

Condition after mixing is given from specific volume.

$$\begin{aligned}
&\text{Specific volume} = \tfrac{2}{3}\tfrac{0}{2}\tfrac{1}{5} = 6.18;\\
&\text{Pressure} \quad = 47.75 \text{ lbs. per sq.in.};\\
&\text{Temperature} \quad = 35^\circ \text{ F. (superheated } 15^\circ);\\
&\text{Heat content} \quad = 553.1;\\
&\text{Entropy} \quad = 1.153.
\end{aligned}$$

After compression to 181.8 the conditions are:

$$\begin{aligned}
&\text{Entropy} \quad = 1.153;\\
&\text{Heat content} = 638.0.
\end{aligned}$$

$$\begin{aligned}
\textit{Work of compression} &= (M_1+M_2)(i_2-i_1)+A(p_1-p_0)v\\
&= 32.5[638-553.1]+\tfrac{1}{778}\,144(47.75-29.3)201\\
&= 3453 \text{ B.t.u. per min.}
\end{aligned}$$

$$\text{I.H.P. of motor} = \frac{3453}{42.42\times 0.75} = 108.5.$$

The power required in Problem 21 was 102, so that this is not of any advantage on account of the lower back pressure. If, however, the load could be divided so that a smaller tonnage would be taken, at the higher pressure then there might be some economy. If 18 tons are used at the higher pressure the results are better.

$$M_1 = \frac{62.2\times 199.2}{492.3} = 25.1 \text{ lbs.};$$

$$M_2 = \frac{18\times 199.2}{495.9} = 7.2 \text{ lbs.}$$

Volume of cylinder for low pressure $= 25.1 \times 8.02 = 201$.

Volume of cylinder for high pressure $= 25.1 \times 6.32 + 7.2 \times 5.92 = 201.2$.

This checks and the system will operate.

Specific volume of mixture at 47.75 lbs. $= \frac{201}{32.4} = 6.21$.

Temperature $= 35°$ F.
Heat content $= 553.0$;
Entropy $= 1.153$.

After compression to 181.8 at entropy 1.153 the heat content is 638.

$$\text{I.H.P.} = \frac{1}{42.42 \times 0.75}[32.3(638-553) + \tfrac{1}{778}\, 144(47.75-34.6)201]$$
$$= 100.2.$$

This means a **saving of 2%** over the simple arrangement. With other conditions this saving may be greater.

Problem 23. Find the **quantity of water** at 60° F. which **gives the most economic results** if water for condensing is raised 100 feet from a stream. Use data in Chapter IX to fix costs. Assume temperature in coils to be 5° F.

(a) Find **cost of producing 1 ton** of refrigeration if temperature of condensation has different values by methods below.

DATA COMPUTED FOR DIFFERENT QUANTITIES OF WATER

t of condensation	70°	75°	80°	85°	90°	95°	105°
p of condensation	129.2	141.1	153.9	167.4	181.8	197.3	231.2
s of compression	1.1637	.....	.....	.....	.....	.....	.....
i at beginning of comp.	539.9	.....	.....	.....	.....	.....	.....
i at end of comp.	621.3	627.1	633.5	639.6	645.2	651.9	666.0
i of liquid at 68° F.	39.9						
t of water	65° F.	68° F.	70° F.	75° F.	80° F.	85° F.	95° F.
I.H.P.	1.02	1.10	1.18	1.25	1.32	1.41	1.59
Gallons per minute	5.56	3.52	2.83	1.91	1.44	1.07	0.85
I.H.P. pump	0.187	0.118	0.096	0.064	0.049	0.036	0.029
Steam per hour for engine	24.5	26.4	28.3	30.0	31.5	33.8	38.0
Steam per hour for pump	28	17.8	14.4	9.6	7.8	5.4	4.3
Total steam per hour	53.5	44.2	42.7	39.6	39.3	39.2	42.3
Cost of coal and labor in cts.	1.07	0.88	0.85	0.79	0.79	0.78	0.85
Fixed charges on engine	0.038	0.041	0.044	0.046	0.050	0.052	0.059
Fixed charges on pump	0.105	0.066	0.054	0.036	0.027	0.020	0.016
Fixed charges on condenser	0.014	0.009	0.007	0.006	0.005	0.005	0.005
Total in cts per ton per hr.	1.227	0.989	0.955	0.878	0.872	0.857	0.930

The method of computing is given as follows:

$$M = \frac{199.2}{539.9 - 39.9} = 0.398 \text{ lb. per min.}$$

$$I.H.P. = \frac{0.398 \times (621.3 - 539.9)}{42.42 \times 0.75} = 102.$$

$$\text{Gal per min.} = \frac{0.398(621.3 - 39.9)}{(65 - 60)62.4 \times 231} \times 17.28 = 5.56.$$

$$I.H.P.\ pump = \frac{5.56 \times 231 \times 62.4 \times 100}{1728 \times 33{,}000 \times 0.75} = 0.187.$$

Steam consumption of engine on compressor, 24 lbs. per I.H.P. hr.

Steam consumption of pump, 150 lbs per I.H.P. hr.

Steam per hr. $= 24 \times 1.02 = 24.5$.

Steam per hr. $= 150 \times 0.187 = 28$.

Cost of buckwheat coal, $3.25 per ton. Cost of firing, 40 cts. per ton. Efficiency of boiler, 65%. Temperature of feed, 200° F., pressure of steam 125 lbs. a bs.

Cost of coal per 1000 lbs. of dry steam

$$= \frac{1000(i - q_o) \times \text{cost per ton}}{\text{Heat per lb.} \times \text{eff.} \times 2240}.$$

$$= \frac{1000[1192.0 - 167.95] \times 365}{12{,}800 \times 0.65 \times 2240} = 20 \text{ cts.}$$

$$\text{Cost of coal and labor} = \frac{53.5}{1000} \times 20 = 1.07 \text{ cts.}$$

Cost of fixed charges on compressor engine of 100 H.P. size based on 15% allowance and 8000 hours of use

$$= \frac{\$20.00 \times 1.02 \times 0.15}{8000} = 0.038 \text{ cts.}$$

There is **no allowance** for fixed charges **on compressor,** as compressor size is the same in all of these cases.

$$\textit{Fixed charges on pump} = \frac{\$300.00 \times 0.15 \times 0.187}{8000} = .105 \text{ cts.}$$

$$\textit{Condenser surface} = \frac{0.398(i_2 - i_5) \times 60}{100 \times \Delta t}$$

$$= \frac{0.398 \times (621.3 - 39.9) \times 60}{100 \times 7.5} = 18.6 \text{ sq.ft.}$$

Cost of condenser at 40 cts. per sq.ft and 15% for depreciation, taxes, etc., and 8000 hours $= \frac{18.6 \times 40 \times 0.15}{8000} = 0.014.$

From the total of cost it is seen that 1.07 gallons per minute is the most economical rate. If now instead of having **water** free from a stream it must be **purchased at 3 cents** per 1000 gallons, the sums above are increased giving the following table:

Gallons per minute.....	5.56	3.52	2.83	1.91	1.44	1.07	0.85
Cost for water free.....	1.227	0.989	0.955	0.878	0.872	0.857	0.930
Cost of water.....	1.000	0.632	0.510	0.344	0.259	0.193	0.153
Total cost with water.....	2.227	1.521	1.465	1.222	1.131	1.050	1.083

$$\text{Cost of water} = \frac{5.56}{1000} \times 60 \times 3 = 1.00.$$

The result is the same as before, although, if these results are plotted into a curve, the most economical rate will be found higher. At 6 cents per 1000 gallons the total cost at 1.07 gallons would be 1.243 cts. against 1.236 at 0.85 gallon; showing that at this cost for water, the cost of water would offset the additional cost of power.

Problem 24. Find the **size of cooling tower** to cool the water required in Problem 22 for 80.2 tons of refrigeration with 102 H.P. and a steam consumption of 25 lbs. of steam per horse-power hour. The water is to be cooled from 95 to 60° F. in 70° weather, with the wet bulb temperature of 60° F.

Amount of water from ammonia condenser $= 1.07 \times 80.2 = 85.8$ gallons per min.

Amount of water from steam condenser

$$=\frac{102\times25\times1000}{35\times60\times62.4}\times\frac{1728}{231}=143.0.$$

Total water per minute to tower = 228.8 gallons = 1910 lbs.

Relative humidity from Fig. 92	0.49
Relative humidity at discharge	1.00
Temperature of air at entrance	70° F.
Temperature of air at discharge	95° F.
Temperature of water at entrance	95° F.
Temperature of water at discharge	60° F.
Barometric pressure	14.7
Assume volume of air at entrance	1 cu.ft.

Weight of moisture at entrance = 0.001153 × 0.49 = 0.000564.
Volume of air at discharge

$$=\frac{144(14.7-0.49\times0.3628)}{460+70}\times\frac{(460+95)}{(14.7-0.815)144}=1.095.$$

Moisture in air leaving = 1.095 × 0.002474 = 0.00271 lb.
Moisture absorbed = 0.00271 − 0.00056 = 0.00215 lb.
Assume water entering when 1 cu.ft. of air enters is equal to m''.
Energy Entering:
With water, $m''\times63.01$ $=63.01\ m''$;

With air, $\frac{1.4}{0.4}\times\frac{144(14.7-0.49\times0.3628)}{778}$ $=9.42$;

With moisture, $0.000565[1081.5+(70-50)0.6]=0.62$.

Energy Leaving:
With water, $(m''-0.00215)(28.08)=28.08m''-0.06$.

With air, $\frac{1.4}{0.4}\times\frac{144(14.7-0.815)1.095}{778}=9.86$.

With moisture, $0.00271 \times 1102.3 = 2.99$.

Equating:

$$63.01\ m'' + 9.42 + 0.62 = 28.08\ m'' - 0.06 + 9.86 + 2.99.$$

$$\frac{1}{m''} = 12.7$$

12.7 cu.ft. of air must be taken in per lb. of water entering.

$12.7 \times 0.00215 = 0.0273$ lb. moisture absorbed per pound entering.

Total air per minute $= 12.7 \times 1910 = 24{,}200$ cu.ft. per min. With a velocity of 700 ft. per second, this would require a cross-sectional area of

$$\frac{24{,}200}{700} = 35 \text{ sq.ft.} \quad (5' \times 7'.)$$

An atmospheric tower would require

$$228 \times 1 = 228 \text{ sq.ft.} \quad (15 \times 15'.)$$

A cooling pond for this plant would contain

$$228 \times 70 = 15{,}960 \text{ sq.ft.} \quad (160 \times 100'.)$$

The basin for **spray nozzles** would contain

$$228 \times 2 = 556 \text{ sq.ft.}$$

There would be a set of four $2\frac{1}{2}$-in. nozzles, as each would care for about 70 gallons. Two sets would be required for 35° cooling.

Problem 25. Find the **amount of refrigeration, surfaces for brine cooler, condenser and bunker,** and **fan size for the air conditioner** for a 450-ton furnace (450 tons per day), when the air is at 90° F. and the wet bulb shows 85° F.

(*a*) *Refrigeration:*

Relative humidity (Fig. 93)..... 0.82
Partial vapor pressure $= 0.82 \times .698$.. 0.57
Temperature of air leaving..... 34° F.
Assume air required per minute..... 40,000 cu.ft.

$$\text{Volume of air leaving} = 40{,}000 \times \frac{144(14.7 - 0.57)}{(460 + 90)} \times \frac{(460 + 34)}{144(14.7 - 0.0961)} = 34{,}900 \text{ cu.ft.}$$

Weight of air entering $= \frac{40{,}000 \times 144 \times (14.13)}{53.35 \times 550} = 2790$ lbs.

Weight of moisture entering $= 40{,}000 \times 0.82 \times 0.002137$
$= 70$ lbs.

Weight of moisture leaving $= 34{,}900 \times 0.000327 = 11.4$ lbs.

Water condensed per minute $= 58.6$ lbs.

Water condensed per day $\frac{58.6 \times 60 \times 24 \times 1728}{62.4 \times 231} = 10{,}100$ gal.

Entering:

Energy in air at entrance above 32° F.
$= 0.24 \times 2790(90 - 32) =$ 38,900

Energy in moisture at entrance above 32° F.
$= 70 \times [1097.3 + (90 - 34)0.6] =$ 77,063

Total 115,963

Leaving:

Energy in air at exit above 32° F.
$= 0.24 \times 2790(34 - 32 =$ 1335

Energy in moisture at exit above 32° F.
$= 11.4 \times 1074 =$ 12,250

Energy in water at exit above 32° F. $= 58.6 \times 2.01 =$ 118

Total 13,703

Heat removed, $115{,}963 - 13{,}703 = 102{,}260$ B.t.u. per min

Tons of refrigeration for air alone $= \frac{102{,}260}{199.2} = 513$ tons.

(*b*) *Surfaces required:*

Air temperature entering, 90° F.;
Air temperature leaving, 34° F.;
Brine temperature entering, 24° F.;
Brine temperature leaving, 39° F.

Mean $\Delta T = \frac{(90 - 39) - (34 - 24)}{\log_e \frac{51}{10}} = \frac{41}{2.12} = 19.4.$

Assume velocity of air 900 ft. per min.

$$K = 2.2\sqrt{15} = 8.5,$$

$$F = \frac{102{,}260 \times 60}{19.4 \times 8.5} = 38{,}200 \text{ sq.ft.}$$

Use 2-in. pipes, 20 ft. long. Pipe surface $= 20 \times \frac{1}{1.608} = 12.45$.

Number of pipes $= \frac{38{,}200}{12.45} = 3060$ pipes.

If sections are made up of sections 25 pipes high and 3 sections above each other, the number of rows will be

$$\text{Rows} = \frac{3060}{3 \times 25} = 41.$$

Area for air $= \frac{40{,}000}{900} = 44.4$ sq.ft.

Width between rows of tube $= \frac{44.4}{41 \times 20} \times 12 = 0.65''$.

Total length $= \frac{41 \times (2.375 + 0.65) + 0.65}{12} = 12'\ 4\frac{3}{4}''$.

If three division walls or plates are placed in bunker, this may be made 13 ft. 0 in. long.

The width of bunker will be 30 ft. to allow 5 ft. at each end.

The height will be

$$75(4'' \text{ centers}) + 3 \times 6'' = 26'\ 6''.$$

The **heat loss from bunker** with 3-in. cork insulation on 12-in. brick will be

$$2340 \times \left[90 - \frac{90+34}{2}\right] \times 0.07 = 4580 \text{ B.t.u.}$$

($K = 0.07$, p. 211).

Tonnage in radiation $= \frac{4580}{199.2} = 23$ tons.

Tonnage in brine $= 513 + 23 = 536$ tons.

Brine coils:

Assume ammonia at 9°.

$$\text{Mean } \Delta T = \frac{(39-9)-(24-9)}{\log_e \frac{30}{15}} = \frac{15}{0.692} = 21.7°.$$

Assume velocity of 5 ft. per sec. From Fig. 95, $K = 137$.

$$F = \frac{536 \times 199.2 \times 60}{137 \times 21.7} = 2145 \text{ sq.ft.}$$

Using $1\frac{1}{2}$-in. pipe 20 ft. long and 10 high for one coil, the surface per coil will be

$$200 \times \frac{1}{2.01} = 99.5 \text{ sq.ft.},$$

$$\text{Number of coils } \frac{2145}{99.5} = 22 \text{ coils.}$$

With no circulation in pipe the rules on Fig. 189 would require 29,315 sq.ft., but in this rule the temperature difference is small.

Suppose brine tank is $20 \times 11 \times 8$ ft. The surface will be 936 sq.ft. and the heat loss will be

$$936 \times 0.07 \times \left[90 - \frac{39+24}{2}\right] = 3800.$$

$$\textit{Tons of Radiation} = \frac{3800}{199.2} = 19 \text{ tons.}$$

Total tonnage $= 536 + 19 = 555$ tons.

Condenser surface $= 555 \times 40 = 22{,}200$ sq.ft. If the ammonia were in condition of Problem 21, the surface would be

$$555 \times 7 = 3885 \text{ sq.ft.}$$

This would require 12 stands of 2- and 3-in. double pipe condensers, each 10 high and 20 ft. long.

$$\frac{3885}{20 \times 10 \times 1.608} = 12.1.$$

(*d*) *Fan Size.* Pressure in actual plants (p. 321) = 1.2 oz.
Use Buffalo conoidal type.
Pressure 1.2 oz.

Equivalent volume at 2 oz. $= 40{,}000\sqrt{\frac{2}{1.2}} = 51{,}600$ cu.ft. per min.

Use No. 130 fan;

$$\text{R.P.M.} = 291\sqrt{\frac{1.2}{2}} = 226;$$

$$\text{Actual quantity} = 64{,}700\sqrt{\frac{1.2}{2}} = 50{,}000;$$

$$\text{Power} = 58.6\ \frac{1.2}{2}\ \frac{50{,}000}{64{,}700} = 27.1 \text{ H.P.}$$

The fan is larger than required, but it is the nearest that can be obtained from table. The fan would be 129 in. high, 6 ft. 6 in. wide and 109½ in. long.

Problem 26. Find the amount of **refrigeration** and power **to operate** a **water-cooling system** to supply 1000 men in a plant when the length of the circuit is 5100 ft. arranged in parallel circuits 1700 feet long with 20 elbows. The water is 75° in 90° weather.

Men on 1 shift..... 600
Quantity of water..... $600 \times \frac{1}{4} = 150$ gals. per hr.
$= 50$ gals. per hr. in circuit
$= 412$ lbs.
Number of fountains..... $\frac{600}{30} = 20$

Length of pipe at each fountain, 20 ft.
Total length of 1 circuit, $1700 + \frac{20}{3} \times 20 = 1840$ ft.
Elbows in 1 circuit $= 4 \times 7 + 20 = 48$.

(*a*) *Refrigeration:*

Mean temperature of water, 50° F.;
Drop in line temperature, 5°—52.5 to 47.5° F.

Using ice water covering and assuming $1\frac{1}{2}$-in. pipe, the heat loss is

$$Q = 1840 \times 0.23(90-50) = 16{,}900.$$

Weight of water to care for radiation with 5° fall

$$= \frac{16{,}900}{5} = 3380 \text{ lbs. per hr.}$$

Area required to give 3-ft. velocity (30), p. 335.

$$F_p \times 3 \times 3600 \times 62.4 = 412 + 3380;$$

$$F_p = 0.00562 = 0.81 \text{ sq.in.}$$

Use $1\frac{1}{4}$-in. pipe.

For $1\frac{1}{2}$-in. pipe

$$\text{Velocity} = \frac{3792 \times 144}{3600 \times 62.4 \times 2.036} = 1.2 \text{ ft. per sec.}$$

Total refrigeration in pipe and water

$$= \frac{16{,}900 \times 3 + 1236[75-50]}{199.2 \times 60} = 6.83 \text{ tons.}$$

Loss in two storage tanks of 6 ft. diameter, 10 ft. high

$$= 220 \times 0.07 \times 40 = 617 \text{ B.t.u. per hr.} = 0.05 \text{ ton.}$$

Total tonnage = 6.88 tons.

(*b*) *Power required:*

$$\text{Head loss} = \left[\frac{0.025}{\left(\frac{1\frac{1}{2}}{12} \times 1.2\right)^{1/4}} \frac{1840}{\frac{1\frac{1}{2}}{12}} + 48 \times 0.2 \times \frac{4.5}{1.5}\right] \frac{(1.2)^2}{64.3} = 13.8 \text{ ft.}$$

$$\text{Power} = \frac{[3 \times 3380 + 1236]13.8}{33{,}000 \times 60 \times .60} = 0.115 \text{ H.P.}$$

Use $\frac{1}{4}$ H.P.

Problem 27. Using data from test of Feb. 5, 1908 (p. 400), reduce refrigerating effect.

(*a*) From brine:

Weight of brine per revolution of pump . . . 41.15 lbs.
Revolutions of brine pump in 15 min. 419

$$\text{Weight of brine per minute, } \frac{419 \times 41.15}{15} = 1150.$$

Temperature of brine at inlet to cooler..... 25.11° F.
Temperature of brine at outlet from cooler. 14.81° F.
Specific heat of brine..... 0.678

Heat removed per minute
$$= 1150 \times (25.11 - 14.81) \times 0.678 = 8049.$$

$$\text{Tons of refrigeration} = \frac{8049}{199.2} = 40.2.$$

I.H.P..... 55.83

$$\text{I.H.P. per ton} \ldots\ldots \frac{55.83}{40.2} = 1.39$$

(*b*) From ammonia:

Mean discharge temperature..... 146.4
Discharge pressure by gauge..... 185.06
Barometer..... 15.01
Absolute pressure..... 200.07
Temperature of saturation..... 95.9
Heat content at 185.06 lb. and 146.4° F. 595.9
Temperature at expansion valve..... 58.91° F.
Heat content of liquid at 58.91° F..... 29.8
Temperature in suction..... 17.80
Pressure of suction..... 20.45
Barometer..... 15.01

Absolute pressure..... 35.46
Temperature of saturation..... 6.5° F.
Heat content at 35.46 lbs. and 17.8° F... 547

$$\text{Ammonia per minute } \frac{236.6}{15} = 15.8 \text{ lbs.}$$

$$\text{Refrigeration} = 15.8[547 - 29.8] = 8180.$$

$$\text{Tons of refrigeration, } \frac{8180}{199.2} = 41.2.$$

This is slightly greater than the brine result.

$$\text{Cooling} = 15.8(595.9 - 29.8) = 8960 \text{ B.t.u. per min.}$$

Problem 28. Check data from test of Westinghouse-Leblanc machine.

$$\text{Refrigeration} = \frac{19826}{60} \times 0.833 \times (18.40 - 15.00) = 935.8.$$

$$\text{Tons of refrigeration, } \frac{935.8}{199.2} = 4.69.$$

INDEX

PAGE

A

Absorber 32, 137
, tubular 37
Absorption machine 31, 79
system 49
Accumulator 279
Adiabatic, construction of 114
After cooling 69
Air 105
blower 295
circulation 245
compressors, cost of 351
cooling 318
of churches, hotels, auditoriums 322
drying 319
drying design 444
for cooling tower 176
leakage, heat of 213
lift pump 298
machine advantage 21
operation 25
work of compression 20
pump 283
quantity 248, 254
refrigerating machines 18
required for raw water ice 295
required for room 420
supply header 295
system closed 19
open 20
velocity 248
Allen 24
Allen dense air machine 19
Ammonia compressors, cost of 351
evaporated, in coil 420
main, size of 425
required 257
Amount of refrigeration for ice making 310
Analyzer 33

PAGE
Apples ... 227
Applications of refrigeration ... 312
Aqua ammonia ... 31
, partial pressure ... 80
, specific heat ... 83
, specific gravity ... 83
, temperature of boiling ... 80
Arctic machine ... 125, 126
Audiffern-Singrun, machine ... 131
Auditorium air cooling ... 322
Automatic refrigeration ... 263
Automobiles, use of ... 311

B

Bananas ... 229
Baudelot cooler ... 314
Beal ... 297
Bell-Coleman ... 21
Belting, cost of ... 352
Berthelot ... 81
Bertsch ... 167
Binary refrigeration ... 169
Blast furnace application ... 318
Bohn ice box ... 13
Boilers, cost of, dimensions of ... 348, 363, 365
Boiling-point ... 2
Boyle Union ... 142
Branch tees ... 145
Brewery ... 241, 314
, refrigeration for ... 317
Brine ... 258
, amount of ... 259
cooler ... 160, 163, 164, 306
, forcing ... 259
freezing tank ... 273
, kind of ... 259
pipe and pump, size ... 423
, specific heat ... 258
system ... 27, 245
tank ... 34
tank coil ... 259
velocity determination ... 420
Buildings, cost of ... 343
Bunker ... 247
piping ... 256
room ... 323
surface, determination of ... 421

C

PAGE
Cabbages ... 229
Candling ... 220
Candy ... 312
Cans, cost of ... 357
Can filler ... 274
, ice ... 289
, number of ... 305
required ... 426
surface ... 310
system ... 269
Car, precooling ... 264
refrigerated data ... 381
refrigerator ... 13
Carbondale machine ... 36
Carbon dioxide machine ... 31, 105, 128
properties ... 390, 391, 392
Carpentering, cost of ... 344
Carré ... 37
Carré Machines ... 4
Carrier ... 51
Carrier's chart ... 175
Celery ... 229
Cement wall ... 201
Central refrigerating plant ... 260
station load ... 262
Characteristic equation ... 62, 64
Cheese ... 223
Chemical work ... 337
Chocolate, specific heat ... 313
Church air cooling ... 322
Cleanliness of plants ... 303
Clearance ... 433
effect ... 44, 71, 433
factor ... 46
Clothing ... 229
Closed air system ... 19
Coefficient for brine pipes ... 306
of transmission of pipes ... 255
Coil, cooling ... 18
, cost of ... 352, 353
, data for ... 352, 353
surface ... 310
, amount of ... 254
, required ... 427
, testing ... 123, 124
Coke filter ... 300

PAGE
Cold storage ... 217
, average length of time ... 219
for brewery ... 241
for florists ... 231
for hotels ... 238
for markets ... 232
for packing houses ... 239
for ships ... 240
heat loss ... 243
laws ... 217
products, value of ... 217
with ice ... 241
warehouses ... 5
Cole. I. & W ... 26
Comparison of thermometers ... 395
Complete absorption, heat of ... 81
dilution, heat of ... 81
Compressed air machines ... 4
Compression, dry ... 68
, machines ... 4
refrigerating machine ... 26
, wet ... 68
Compressor air, size required ... 431
, cost of ... 351
, ammonia, cost of ... 351
, size required ... 432
, arctic ... 125
, De la Vergne ... 29, 116, 118
, dimensions of ... 358, 360, 361
, exhausting ... 125
, Frick ... 28, 122
, power required ... 432
, single acting ... 27
, York ... 110
, aqua ammonia ... 31
Condenser ... 31, 32, 149
Block, cost of ... 352, 353
, data for ... 352, 353
, De La Vergne ... 154
, design ... 436
, double pipe ... 37
, exhausting ... 123
, flanged ... 152
, Philadelphia ... 157
, oval flask steam ... 165
, screwed ... 153
, Shipley ... 158

PAGE
Condenser, submerged 155
, supports 159
, surface steam 165
, welded 151
Conduction 182, 183
Conduits 261
Congealer 233
Constant quality 58
vapor weight line 58
volume line 59
of transmission 187
Construction warehouses 231
Cool brine system 5
Cooler for sweet water 241
Cooling 46
, by evaporation 1
, by solution 1
, by ice 3
, determination of 399
, drinking water 331
method of 244
pond 172
pond design 178
tower 167, 169
tower design 176, 442
tower test 406
water 374
Cool water coil 32
Cooper system of refrigeration 18
Cork, best thickness 407
board 201
covering 336
loss 334
Correction for hydrometer 395
of thermometer readings 397
Cost of air compressors 351
ammonia compressors 351
belting 352, 353
fan blowers 352
boilers 348
buildings 343
cans 357
carpentering 344
coils 352, 353
condensers 352, 353
distilling apparatus 357
electric generator 350

PAGE
Cost of electric motors 350
engines 349
excavations 340
floors 345
gas engines 350
ice, natural 384
, manufactured 384
storage 309
insulation 345, 346
land 343
lumber 344
machinery 347
masonry 344
millwork 345, 395
miscellaneous apparatus 357
operating 376, 377, 378, 379
partitions 345
painting 345
plumbing 345
pipe and fittings 354
pipe coverings 346
pipe for storage 357
plant, initial 376, 377, 378, 379
plumbing, initial 345
producers 348
pumps 351
receiver 353
roadway 345
roofing 345
separators 353
space 409
storage 215, 375
supplies 357, 374
switchboard 350
water 374
Counter current flow 47
Curves, construction of 115
Cream 225
Creamery refrigerators 339
Crosses 145
Curve of ice consumption 308
Cushionhead 109
Cycle diagram 66
Cylinders 116, 120, 122, 434
Cylinder expansion 19
head 29
operation of 123

D

PAGE
Dairy refrigeration 339
Data for coils and condensers 352, 353
engine 349
gas engine 350
ice cream 382
ice delivery 380
ice storage plant 310
pipes 355, 356
pumps, ... 351
rinks 382
turbines 349
warehouse 383
Deepwell pump 298
Dehydrator 33, 34
De la Vergne 116, 118, 120
machine 30
freezing tank 273
Dense air machine 57
Density of salt 258
Deodorizer 284
Depreciation 206, 379
Design of pistons 435
piston rod 435
Determination from Le Blanc machine test 451
amount of ammonia and air 420
amount of water for condenser and power 432
of best quantity of water 440
of best thickness of cork 407
brine pipe and pump 423
bunker surface 421
coefficient of wall 408
coil surface 427
condenser surface 436
cooling 399
cost of pumping 429
data for air drying 444
fan and power 422
heat loss through walls 413
ice storage 428
length of pipe 419
multiple effect installation 438
number of cans 426
pipe sizes 435
plate plant 426
radiation 418
refrigeration 396, 398, 428

PAGE
Determination refrigerating effect test 449
size of air compressor 431
size ammonia main 425
size of cooling tower 442
of size of evaporator 430
size of filter 431
size of freezing tank 425
specific heat 398
storage space required 410
value of ice storage 428
of valve area 435
velocity of brine 420
water-cooling system 448
Dexter system of refrigeration 11
Diagram of cycle 66
Dickinson 16, 258, 308
Diffuser 167
Dimensions of boilers 365
compressors 358, 360, 361
Dimensions of engines 362
generators 368
motors 369
producers 367
turbo-generators 363
Direct-expansion system 5, 27, 245
Displacement 70
Distilled water 79
Distilling apparatus 280
, cost of 357
Distribution of air 246
Door, heat leakage 416
construction 208
Drinking water, cooling 331
computations 335
for hotels 338
Dry bulb thermometers 50
compression 68
Drying air 319
Duct, size of 248
Dump, ice 290
Dust preventing 222
Dynamic pressure 250

E

Efficiency apparatus 347
Eggs 220
, candling 220

PAGE
Eggs, cracking ... 221
, temperatures of storage ... 222
, weight ... 220
Elbows ... 143
Electric generators, cost of ... 350
motors ... 115
welding ... 140
Elevator ... 238
Engines, cost of ... 349
, data ... 349
, dimensions of ... 362
, steam ... 115
Equivalent speed ... 252
volume ... 252
Ethyl alcohol ... 105
Evaporating surface, effect of varying ... 306
Evaporation, heat of ... 2
Evaporator, refrigeration by ... 26
, design of ... 283, 302, 307, 430
Excavation costs ... 244
Exchanger ... 34, 137
Expander ... 66
Expansion coils ... 31
, storage of ammonia in ... 123
for freezing ... 306
joint ... 262
valve ... 31, 289

F

Fan blowers, cost of and dimensions ... 352, 353
data ... 251
, size and power ... 422
Fermenting tank ... 241
tub ... 315
Filter ... 276, 286
size required ... 431
Fish ... 225
Flange union ... 141
Flooded system ... 288
Floor construction ... 207
insulation ... 204
costs ... 345
Fore cooler ... 284
Freezing by evaporation ... 37
, coils for ... 306
, tanks ... 303
, time of ... 304

PAGE
Freezing tank 273, 288, 303
, size 425
Frick Co. 27
machine 28, 122, 123, 124, 275
Friction effect 44
loss 249
Fruit 227
Fuels 348
Furs 229
Fusion, heat of 2, 215, 308

G

Gauge board 29
Gas Engine 115
, cost of 350
, data 350
Gayley 319
Generator 32, 134
dimensions 368
George 258
Gobert system 329
Goods, heat from 416
Gorrie 21
Grapes 228
Grease separator 285

H

Hall, J. & E 26
Hampson 340
Hangers 146
Hart cooling tower 171
Haslam & Co 26
Haynes 38
Head cushion 109
Headers 273
Heads 117
, false 119
, spherical 121
Heat content 59
-entropy diagram 61
for breweries 317
from door leakage 416
goods 416
men 416
loss from pipe 334
in cold storage 243
per year 408

PAGE
Heat of air leakage ... 213
complete absorption ... 81
partial absorption ... 81
complete dilution ... 81
fusion ... 2, 215, 308
lights ... 214
machines ... 214
persons ... 213
solution ... 3
of salt ... 14
vaporization ... 2
transfer ... 182
through walls ... 190
transmission ... 184
Helmets ... 179
Hoofnagle ... 38
Hoists ... 290
Horse-power to drive ... 47
Hotel air cooling ... 322
boxes ... 238
Hydraulic radius ... 249
Hydrometer, correction for ... 295
Hygrometer ... 50

I

Ice, amount of refrigeration ... 310
and salt mixture ... 223
, artificial ... 5
, manufactured ... 5
, can ... 289
cold storage ... 241
cream data ... 382
cream freezer ... 326
consumption, curve of ... 308
, cooling by ... 3
, cost of storage ... 309
, delivery data ... 380
, distribution ... 310
, dump ... 290
, heat of fusion ... 308
, latent heat of fusion ... 41
making ... 265
, absorption system ... 297
, passenger car ... 382
plant, Frick ... 275
, York ... 270
saw ... 279

PAGE
Ice storage amount and economy ... 428
plant data ... 310
room ... 276
tank insulation ... 208
Inclined coordinates ... 62
Incomplete expansion ... 48
effect ... 48
Indicator cards ... 41
, from vapor machines ... 64
, use of ... 113
valve ... 29
Indirect system ... 246
Injecting liquid, effect of ... 119
Insulation ... 190, 201
, amount of ... 244
, cost of ... 345, 346
, experimental determination of value ... 212
values of K ... 211
Insurance ... 206, 379
Interchanger ... 33, 37
Interest ... 379
Interstate Commerce Rules ... 181

J

Jacket ... 126
, effect of ... 112
Jackson system of refrigeration ... 9
Johns–Manville Co ... 131

K

Kirk ... 21
Kroechel machine ... 128

L

Labor for plants ... 373
Lagging ... 112
Land cost ... 343
Lantern ... 112
Latent heat ... 2
of fusion of ice ... 41
Laws, cold storage ... 217
Le Blanc ... 39
machine test ... 451
Lightfoot machine ... 19
Lights, heat of ... 214
, heat from ... 416
Lillie evaporator ... 300

PAGE
Linde ... 340
Liquid air ... 340
Liquid-air machines ... 341
Liquid line ... 58
Liquid receiver ... 341
Lith ... 201
Load factor ... 375
Lorenz ... 329
Loss of heat per year ... 408
Low temperature by ice and salt ... 16
Lubrication ... 121
Lucke ... 80
Lumber, costs ... 344

M

Machine, absorption ... 31
, Carbondale ... 36
, York ... 35
Allen dense air ... 19, 57
compressed air ... 4
, heat of ... 214
, Lightfoot ... 19, 57
, test of ... 400–405
Machinery, costs ... 347
Marine compressor ... 129
Market ... 232
Masonry costs ... 344
McCray refrigerator ... 8
Methyl alcohol ... 105
chloride ... 31, 105
Meat ... 223
Melons ... 229
Men, heat from ... 416
, required ... 373
Milk ... 225
Mill work, cost of ... 345
Miscellaneous apparatus ... 357
Motors, dimensions of ... 369
, electric, cost of ... 350
for driving compressors ... 115
Moisture effect ... 50
Mollier ... 80
diagram ... 63
Multiple effect ... 107
, installation ... 438
, Voorhees ... 71

N

PAGE
Nozzle design ... 178
steam ... 167

O

Oil pump ... 119
separator ... 31
spray ... 117
Onions ... 229
Open air system ... 20
Operating costs ... 376, 377, 378, 379
Operation of cylinders ... 123
Oranges ... 228
Osborne ... 16, 308
Ott Jewell ... 297
Oxy-acetylene welding ... 140
Oysters ... 227

P

Packing house ... 239
, leather ... 129
Painting, cost of ... 345
Partial pressure from aqua ammonia ... 80
Partial absorption, heat of ... 81
Partition, cost of ... 345
Passenger car ice ... 382
Patten ... 38
Peaches ... 228
Pears ... 227
Penny ... 169
Performance of plants ... 370, 371, 372
Perkins ... 4
Perman ... 80
Persons, heat from ... 213
Pipe and fittings, cost of ... 354
Pipe coefficients ... 255
covering ... 200, 346
data ... 355, 356
for storage, cost of ... 357
heat loss ... 334
joints ... 142
length, determination of ... 419
lines, bell and spigot ... 262
line, brine ... 262
line, size ... 257
, size at compressor ... 435
, size of ... 308

PAGE
Pipe, suction and discharge 119
Piping 138
arrangement 236
for brine tanks 259
for bunkers 256
for rooms 256
Piston 27, 109, 129
, arctic 126
design 435
rod 121
rod attachment 121
rod design 435
speed 434
Planck 75
Plant cost 376, 377, 378, 379
Plate plant, size of 426
Plate system 269, 276, 297
Plumbing, cost of 345
Plums 228
Poetsch system for shaft sinking 328
Point of boiling 2
fusion 2
melting 2
Pond, cooling, design 178
Poultry 224
Power for deep well 300
plants 370, 371, 372
raw water ice 296
to drive 106
Precooling cars 264
charges 382
Pressure, effect of varying 307
volume diagram 64
Principle of refrigerating machines 5
Problems, absorption machine 83
, air machine 55
, vapor machines 72
, miscellaneous 407–451
Producers, cost of 348
dimensions 367
Properties of ammonia 387–389
of carbon dioxide 390–392
of sulphur dioxide 393–395
Pump, air lift 298
, cost of 351
data 351
, deep well 298

PAGE
Pump, oil ... 119
Pumping, cost of ... 429
Purge valve ... 29

R

Radiation ... 182
, required ... 418
Raw water ... 79, 269
system ... 290
Reboiler ... 274, 282, 284, 285
, vacuum ... 283
Receiver ... 161
Receiver, cost of ... 353
Rectifier ... 33, 34, 136
Reduced pressure ... 3
Refrigerants ... 105
Refrigerating capacity ... 41
effect ... 46
effect, vapor machines ... 70
effect from test ... 449
machine compression ... 26
machines, general principle ... 5
machines, diagram of cycle ... 66
mediums ... 31
plants, cost of ... 347
Refrigeration ... 46
applications of ... 312
automatic ... 263
by chemical process ... 40
ice ... 8
evaporation ... 26
, central station ... 260
, determination of ... 396, 398
for brewery ... 317
creamery ... 339
dairy ... 339
plant ... 428
methods ... 8
Refrigerator ... 8
cars ... 13
household, data ... 384
Relative humidity ... 50
chart ... 175
Repairs ... 379
Return bends ... 31, 144
Return tubular boilers ... 363
dimensions of ... 363

PAGE
Rietschel ... 191
Rinks ... 326
Rink data ... 382
Roadway, cost of ... 345
Roelker ... 24
Roofing, cost of ... 345
Rooms, piping for ... 256
Rooms, temperature of ... 244, 413
Rugs ... 229
Rules for safety ... 180

S

Safety devices ... 179
head ... 29
plate ... 129
Salt, heat of solution ... 14
Saturated ammonia, properties of ... 385–387
Saturation line ... 58
Scale separator ... 31
Separator ... 33, 34, 119, 159
, cost of ... 353
, oil ... 31
scale ... 31
Setting box ... 312
Shaft sinking ... 328
Ship cold storage ... 240
Single acting compressor, advantage of ... 113
Skimmer ... 283
Solution, heat of ... 3
Space, cost of ... 409
, storage, determination of ... 410
Spangler ... 80
Specific heat, determination ... 398
of aqua ammonia ... 83
brine ... 15, 258
chocolate ... 313
ice ... 16
materials ... 215
superheated steam ... 44
vapors ... 64
Speed, equivalent ... 252
Spider ... 109
Spray nozzles ... 172
Stahl ... 38
Static pressure ... 250
Steam for plants ... 371
washer ... 300

PAGE
Storage, cost of 215, 375
tank 31, 287
unit for 215
Strainer 119
Strawberries 228
Stuffing box 27, 112, 117, 121, 126
Suction side 29
valve 28
Sulphur dioxide 31, 105
machine 131
properties of 391–393
Supplies, cost of 357, 374
Superheat, degrees of 63
Sweet water cooler 241
Switchboard, cost of 350

T

Taxes 379
Tees 143
Temperature at points on cycle 19
-entropy diagram 57
mean difference 185
of freezing for brine 258
ice and salt mixtures 14
rooms 244, 413
range, effect of 78, 79, 106
Testing coils 124
Test of apparatus 395
cooling tower 406
machines 400–405
Tests 395
Thermit welding 138
Thermodynamics of refrigeration 41
Thermometer comparison 395
correction 397
, use of 169
Thomas spray nozzle 172
Thompson-Joule effect 340
Throttle valve 32
Tilting table 279
Time of freezing 304
storage 215
Tobacco 229
Tomatoes 229
Tripler 340
Triumph ice machine 127
Tub, fermenting 315

PAGE
Turbines, cost of ... 349
data ... 349
Turbo-generators, dimensions ... 363
Twining ... 4
Two cylinders, use of ... 113

U

Ulrich ... 297
Unions ... 141
Unit for storage ... 215

V

Values of K ... 187
Valves ... 121, 148
, cylinder ... 117
, hurricane ... 125
, indicator ... 29
, manipulation of ... 123
, mushroom ... 18
, purge ... 29
, sizes ... 435
, slide ... 18
, Triumph Co ... 127
Vaporization, heat of ... 2
under reduced pressure ... 3
Vapor machine ... 57
pressure ... 51
pension ... 51
Vegetables ... 229
Velocity of air ... 248
pressure ... 250
Ventilating air cooling ... 322
Vogt Co ... 134
Volume equivalent ... 252
Volumetric efficiency ... 70, 434
Voorhees ... 71, 107, 438

W

Wall coefficient, determination of ... 408
constants ... 192
for cold storage ... 243
heat loss ... 413
Warehouse construction ... 231
data ... 383
Water, best quantity of ... 440
, cooling system design ... 448
, cost of ... 374

PAGE
Water, distilled ... 79
, distilled, amount required ... 300
, effect of large quantities in air ... 53
for condensing ... 432
for cooling ... 374
for ice making ... 298
jacket ... 29
jacket, value of ... 29
per pound of aqua ammonia ... 81
, raw ... 79
storage tank ... 273
tank insulation ... 337
tube boiler, dimensions ... 365
Weak liquor cooler ... 137
Welding ... 138
Westinghouse ... 39
Westinghouse—Le Blanc machine ... 167
Wet and dry bulb hygrometer ... 174
Wet bulb thermometer ... 50
Wet compression ... 68
White-wash ... 222
Wood insulation ... 202
Work of compression ... 43
, air machine ... 20
expansion ... 43
with friction ... 44

Y

York ice plant ... 270
machine ... 36, 110, 111

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